CENTRIFUGAL
PUMPING MACHINERY

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CENTRIFUGAL PUMPING MACHINERY

THE THEORY AND PRACTICE OF CENTRIFUGAL AND TURBINE PUMPS

BY

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PREFACE

A WRITER upon centrifugal pumps has said that they defy the mathematician and possess more tricks than a circus mule.

This book has been prepared with the idea of supplying accurate and definite information, which can be used in actual design. The data are based upon experience in design, construction and installation of this type of pumping machinery. It attempts to set forth the underlying principles, which, if properly applied and used, will give the designer enough information to enable him to calculate results with reasonable certainty. Almost all books on the subject in the English language are silent upon the principles which govern the practical designer, and give only empirical formulas which would prove very costly in actual practice.

This book confines itself to material which has been used successfully in practice. The author has himself had charge of almost all of the installations described and they become of interest, therefore, as records of fact.

Necessarily, the installations given are from the practice of Henry R. Worthington, and it has not been deemed wise or necessary to introduce other practice to illustrate the principles which the book seeks primarily to set forth. This is not merely because of the large number of centrifugal and turbine pumps which this company has put out, but because it seems best to confine the work to actual experience and to such installations as have been in service sufficiently long to be beyond the experimental stage.

No attempt has been made to go into the history of the subject, or to treat it from an elementary standpoint, as it is assumed that the reader is familiar with the laws of hydraulics.

HARRISON, NEW JERSEY
April, 1912.

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CENTRIFUGAL PUMPING MACHINERY.

PART I.

CHAPTER I.

GENERAL REMARKS.

LOW- AND HIGH-LIFT PUMPS.

THE methods used for figuring and designing centrifugal pumps are usually regarded as more or less mysterious. This is due to the insufficiency of the data and information available in the English technical literature at the present time. Foreign writers have treated the theoretical side very carefully and thoroughly. Private investigations have been developing and enlarging upon the original theories and showing how they work out in actual practice, but only a few of these results have been placed in the hands of the public.

The centrifugal pump presents many interesting phases which do not appear in any other style of pumping machinery, and these must be understood and their importance appreciated for the intelligent design, operation, or application of the pump. Many of the peculiar features have been demonstrated graphically, and should be carefully studied in Chapter V on Efficiency in this part. Some of these peculiarities afford a convenient classification of centrifugal pumps by characteristics. The present discussion, however, will classify the pumps as low-lift and high-lift, according to the head pumped against.

LOW-LIFT PUMPS.

Although an increase in speed conditions may change a low-head pump to high-head, pumps intended for the latter service are differently constructed, as will be seen. Low-lift pumps have, until recently, been little understood. In design and construction they have been crude and uneconomical, but pumps of this type are now built which secure high economy at both high and low heads. This has been the means of putting them into general use for a great variety of purposes.

This style of pump is generally of the volute type. Under proper conditions it is probably the lightest and cheapest pump that can be used for moderate and large quantities of water. The best conditions are total

heads of from 0 to 150 feet, short and direct suction and delivery pipes, moderate and large quantities of water — never small quantities.

The chief applications for low-lift pumps are: drainage, reclamation, and irrigation work; waterworks where the lift is not high and where low first cost is desirable; pumping into filter beds; sewerage work, dock work, sluicing, leakage in tunnels, circulating water in condensers for power stations, and general water-service pumps for buildings, mills, and heating plants. The method of applying the centrifugal pump to these services, and the conditions demanded for each service, are explained in detail in Part 4, and should be carefully studied by the designer and engineer when considering pumps for such installations.

Centrifugal pumps will deal with very large volumes of water. Several have been installed which handle as much as 130,000 gallons per minute. These pumps can be built with a discharge pipe as large as 72 inches, and with pipe velocities of 8, 10, 12, 14, and 16 feet per second, corresponding to the volume of water. In fact, the amount of water that can be moved is almost unlimited, as there is no difficulty in constructing pumps for these large amounts of water for heads from 0 to 40 feet and with smaller quantities up to 150 feet.

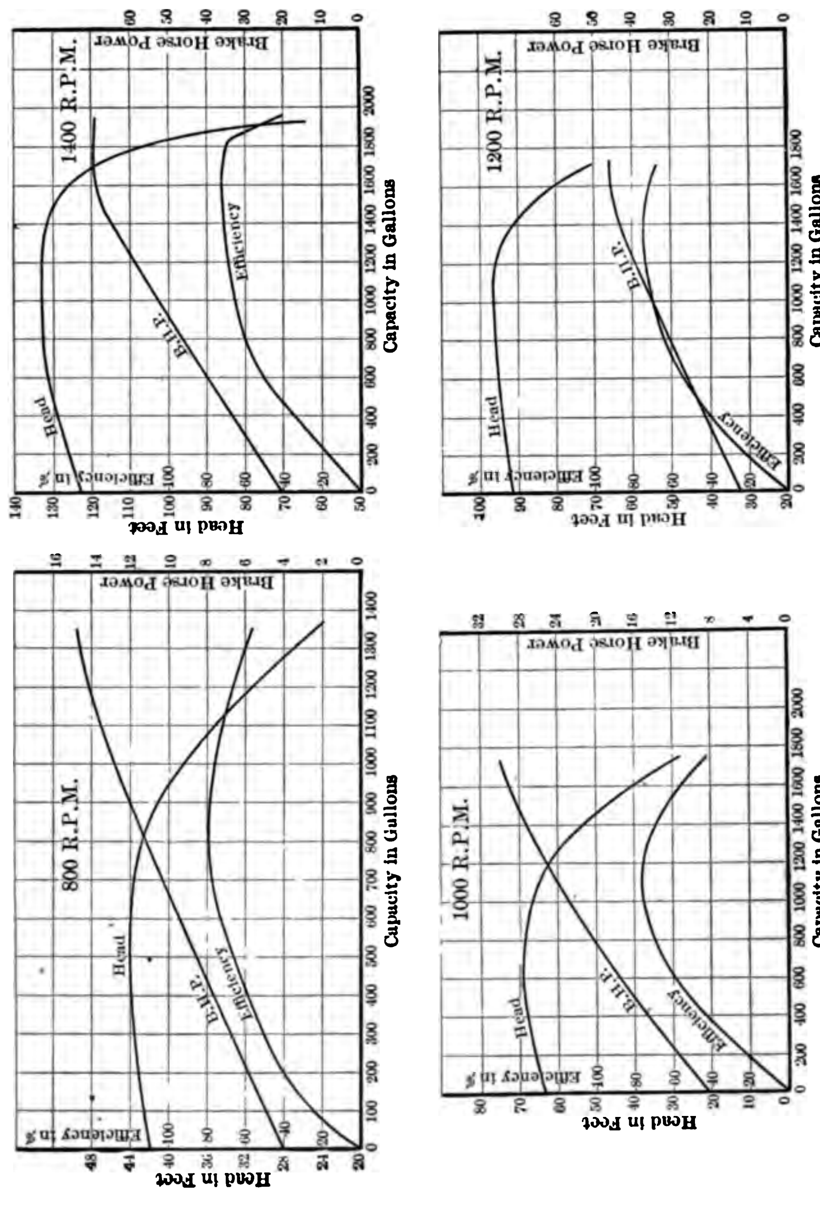
The ordinary form of low-lift volute centrifugal pump is not adapted to high speeds, and for these the multirotor pump or some other type must be adopted. These types are described in the chapter, "Special High-speed Turbine Installations," at the end of Part 3. For very low heads or suction heads only and pumps working on sealed pipes, a combination screw and centrifugal pump or a double-screw pump is employed. This is treated separately in Part 4.

HIGH-LIFT PUMPS.

For heads of more than 150 feet, instead of the volute casing a round casing is used, which resembles that of a water wheel. Because of this resemblance, high-lift pumps are generally known as turbine pumps.

It has only recently become known that the simpler form of centrifugal pumps, with few parts and only one moving piece, could be used for fairly high heads. In the *Transactions* of the Institution of Civil Engineers, London, Volume XXXII, it is stated that an 18-inch pump will work well on a 20-foot lift, and a 36-inch pump on a 30-foot lift. Other writers have stated that the ordinary centrifugal pump has a low efficiency at high heads.

The relation existing between the actual pressure in the pump discharge and the theoretical pressure was not known, and but little attempt had been made to ascertain it. In the design of centrifugal pumps this ratio, called by some authorities the manometric coefficient based on experiments, must be the basis from which to work, together with the percentage of the useful work to that expended in operating the pump.



Figs. 1 and 2. Results of Single Propellers without the Use of Diffusion Vanes.

Practical results have shown that single impellers without diffusion or guide vanes can be made for heads as high as 350 feet, and it is not unusual to find commercial pumps for heads of 150 feet in which the rapidly moving water leaves the impeller with very little loss from shock or eddies. It is of course necessary to have high peripheral velocities in order to obtain such heads. Figs. 1 and 2 give results for single impellers without diffusion vanes for heads from 40 to 135 feet, in the same pump, under different speeds.

Very high heads can be obtained by multistage pumps, which consist of a succession of rotors connected in series through the casing, so that each stage draws its suction from the discharge of the preceding one and so raises the pressure progressively. Pumps of the multistage type are made for heads as high as 200 feet per stage when proper conditions are met as to capacity and speed. In good practice the head per stage varies between 100 to 150 feet, in order to keep down the velocity of the water so that it will not cause trouble from pitting the vanes or producing excessive wear on the impeller and diffusion tips. The velocity of flow in the suction inlets of the pump must also be kept low, or the water may separate from the entrained air and the pump become noisy, due to the high speed of the impellers and imperfect filling.

CHAPTER II.

DIFFUSORS.

In the design and operation of centrifugal pumps it is important that no head should be lost by shocks or abrupt changes of velocity. The ordinary method of taking care of abrupt changes in velocity consists in introducing a whirlpool chamber and volute, stationary or movable diffusors, or stream lines. In low-lift pumps the casing is usually in the form of a volute, with a gradually increasing cross section. In high-lift multistage pumps, each section or stage is fitted with a whirlpool chamber or a device with divided stream passages known as diffusion vanes which divert the water on leaving the wheel from a tangential direction into the proper one for discharge. This whirlpool chamber is usually an Archimedian spiral, to allow the water to move freely and to convert its velocity into pressure with as little friction as possible.

It has been shown by experiment that a pump with a proper whirlpool chamber will work with greater efficiency and against a greater head than one without it. This is easily understood when we consider that the mass of water revolving outside of the wheel has some centrifugal force, which can be added to that produced by the wheel to increase the pumping head. Or, instead of this chamber, a stationary guide-vane chamber, called a diffusor, can be made. Both of these types are intended to utilize the energy of the rotating water as it leaves the wheel, and increase the pumping power. The guide vanes or diffusors must be so designed that the stream lines will not produce too much skin friction due to additional surfaces.

Efforts have been made to have these guide vanes adjustable like those of the water turbine, but this has not proved practical. For the best efficiency, therefore, they should have the shape suited to the water path corresponding to the conditions for which they were designed. The vanes, in some cases, are made as a separate or removable casting, in others as spiral grooves cast in the main casing, or as passages in delivery compartments of the casings. The spaces in the vanes are supposed to have the shape which the water would assume in its passage from the wheel under certain and fixed conditions, but the exact conditions of flow are not known, and as the guides are always less in number than the vanes in the wheel, the water is liable to strike them at the entrances of the ring and thus cause injurious eddies. Diffusion rings can be made without vanes and good efficiencies obtained even for high heads.

Investigation of movable diffusors for high heads, made to rotate freely on the impeller, has also been made, the results of which show that higher efficiency and heads can be obtained. This diffusor was originated by Mr. Barber, March 31, 1896, and by Professor Novak of Austria in 1908. Its purpose is to reduce the friction on the sides of the runners, and to provide ample whirlpool for the diffusor chamber in the pump, doing away with the vanes usually employed.

In the usual construction, the water between the runner or impeller and the stationary chamber has a tendency to churn, causing a waste of energy which increases with the clearance. A properly designed movable diffusor will greatly reduce this loss and thereby increase the efficiency.

CHAPTER III.

BALANCING THRUST.

More or less axial thrust is present in all centrifugal pumps. It is caused by the unbalanced pressure between the impeller and casing, between the impeller and channel or filling-in rings, and also where pressure acts upon unequal surfaces. The water in passing through the wheel alters its velocity and pressure, and in changing from an axial to a radial direction produces a centrifugal force causing a thrust. The vane angles have some influence upon the amount of thrust, dependent upon the velocities at the inner and outer angles of the vane with respect to the impeller. The shape or form of the vanes may also cause a slight thrust under certain conditions. Very little is known definitely of this matter, but experiments have shown that if we assume an impeller with a cross section S, and a velocity V_3 at the hub, the axial thrust will be approximately $3500\,S\,\frac{V_3^2}{g} = T$, where T is expressed in pounds of thrust for each impeller.

Various arrangements of impellers opposed to each other have been tried in order to eliminate this axial thrust. Impellers having two faces exposed, the smaller side to the higher pressure, the larger one to the lower pressure, have been used, and by so doing the thrust has been materially reduced; but, owing to complications introduced, this method is open to more or less criticism. Arrangements with bushing rings, and impellers with balancing holes, will materially aid in securing equal pressure upon the areas within the bushing rings. For the remaining unbalanced thrust a marine-type ball or roller bearing can be used on the outer end of shaft. Hydraulic thrusts of different types have been designed with a view of automatically adjusting themselves to the conditions, and of taking care of the additional thrust that may be produced by future wear at various points where leakage occurs. The most successful of this type, with a revolving steel disk having very close-running surfaces, is known as hydraulic step bearing, and can be operated by pressure from the discharge with very small loss. Another type consists of an internal disk step located either on the discharge side of last impeller or suction side of the first, and is designed to control the thrust automatically. An adjustable ring or collar within the pump, securable so as to control the opposing pressures in the chambers or casings and produce an opposite thrust, has also been designed.

CHAPTER IV.

PRIMING AND FOOT VALVES.

PRIMING.

Before a pump is started all air must be expelled, and it must be filled completely with water. A centrifugal pump running in air cannot create the vacuum necessary to raise the water up to the impeller. Some pumps, particularly those handling hot water or any liquid giving off a vapor,

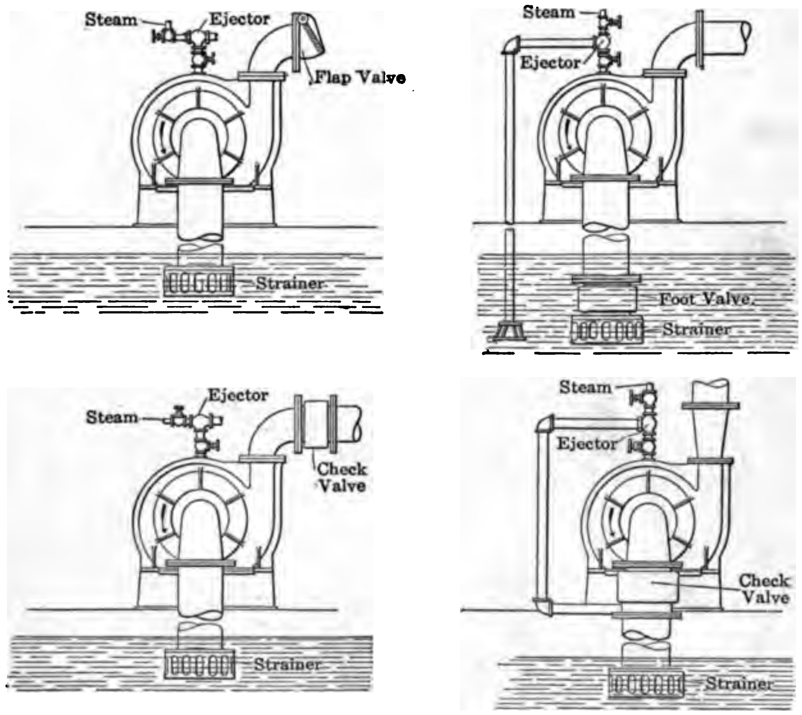


Fig. 2A. Methods of Priming.

should always be so placed that the water will flow to the pump; otherwise the vapor will collect in the pump and it will cease pumping. Pumps so located that water will not flow to them must be primed. Methods of priming are illustrated in Fig. 2A.

There are two principal methods of priming, i.e., by producing a vacuum with an air pump, or by pumping water into the casing; and all the devices

used are different forms of one of these methods. The conditions will determine whether steam, compressed air, or water should be used. A simple method is to have a foot valve on the end of the suction pipe, and to fill the casing through a small hole at the top with city water or other water under pressure. A second method uses the siphon principle. A foot valve is placed at the end of the suction pipe, a gate valve in the discharge close to the outlet where it extends horizontally from bottom of pump, an air cock at the top of the casing, and water is supplied from a connection on the side. When water appears at the top pet cock the pump is primed. After closing the water supply, open the discharge gradually. This initial priming is sufficient for all future starting, pro-

Fig. 2B. Priming Device for Small Centrifugal Pumps.

vided that the foot valve is tight, and that the discharge gate valve is gradually closed as the pump is shut down. The water will then remain in the pump.

For heads of over 30 feet, it is usual to place a check valve in the discharge in order to prevent shock on the pump, and it is customary to place a foot valve on the suction and a by-pass for priming between the discharge, above the check valve, and the suction pipe above the foot valve.

Another method for use with steam or compressed air is to pump water into the casing by an injector. This should be so placed that it is within easy suction lift of the water. Priming without a foot valve, such as is necessary with pumps operating on wells where it is impossible to place one, requires a check valve in the suction pipe close to the pump opening.

An ejector is connected at the top of the casing and its suction pipe tapped into the main suction under the check valve.

A method of priming with a foot valve, and a check valve in the discharge, using an ejector as an exhauster, is also employed. For this method both steam and compressed air are used.

On large engine-driven centrifugal pumps, running condensing, the top of the casing can be connected with the condenser and a sufficient vacuum created for priming. In such installations it is best to fit a glass water gauge on the top of the pump so that the operator may know when the casing is full and prevent the water from going into air pump, particularly if this is of the dry, rotative type.

For small centrifugal pumps, up to and including 12 inches, a specially designed priming device may be used. (See Fig. 2B.) It consists of a fitting or casting placed directly against the suction opening and takes the place of the usual elbow. It is fitted with a small hand pump, and a clapper or foot valve in the main passageway. Water is drawn through the main opening into the hand pump, and forced out through a small check valve into the main pump, the water being retained there by the main check valve in the primer. In order to prevent losses from friction in this type of primer, the openings are made very large.

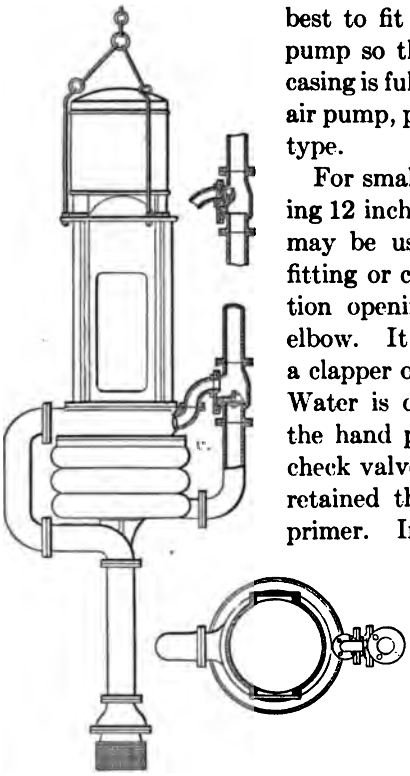


Fig. 3. Automatic Priming Arrangement for Sinking Pumps.

A similar type of primer providing the same features consists of an ordinary hand pump attached to the suction opening and connected with the water supply. A foot valve is required on main suction.

Still another method is to attach a hand air pump onto the pump at any place below the discharge gate valve for exhausting the air between foot valve and discharge valve.

In larger installations, separately driven electric or steam-driven vacuum pumps, operated automatically, are used, particularly where the suction pipes are large and quite long.

In mining installations it is necessary with sinking pumps to use an automatic repriming arrangement, illustrated in Figs. 3 and 4, consisting of a foot valve in the suction and an automatically operated check valve

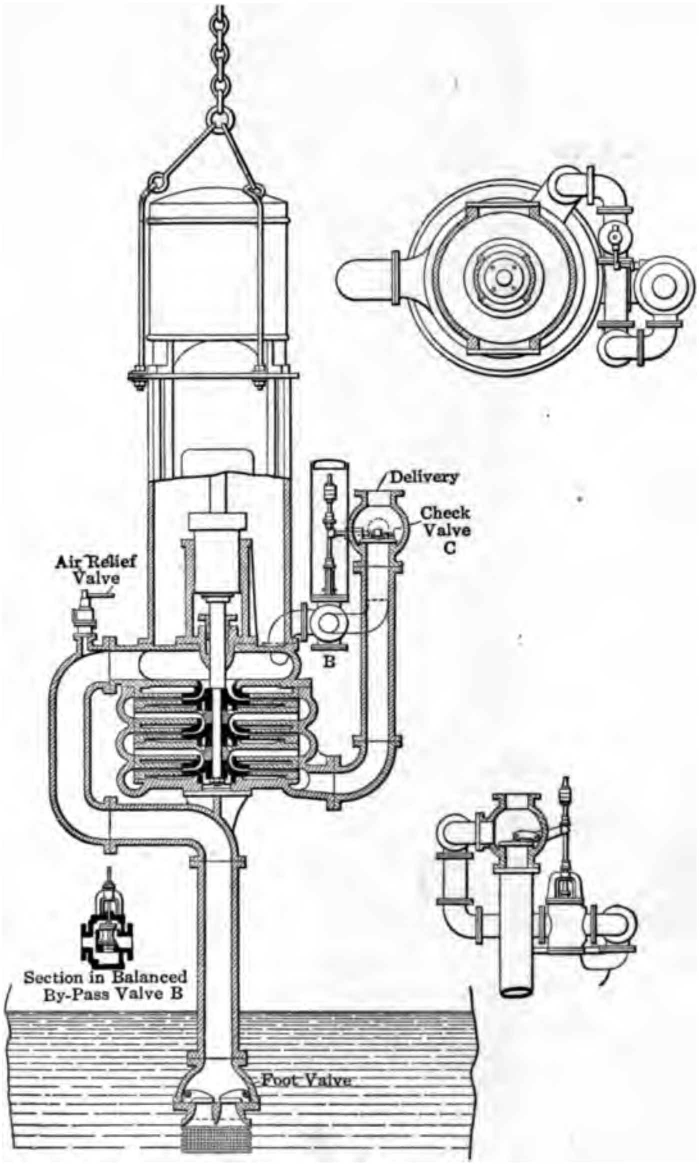


Fig. 4. Automatic Priming Arrangement for Sinking Pumps.

in the discharge, with necessary air-relief valves at the highest point of the entrance pipe to the pump. Its purpose is to discharge the air and to refill the suction pipe. The illustration shows the automatic check valve, which is wide open when pumping, allowing a free discharge, at the same time closing off the connection between the delivery pipe and the entrance to the first impeller. Should the pump stop for any reason, the check valve closes the connection from the last impeller to the column and allows the water in the column pipe to flow through the small pipe and submerge the impellers, priming the entire pump down to the foot valve. On starting up any accumulation of air in the suction pipe will be discharged through the automatic air-relief valve. When all the air has been discharged the air-relief valve closes.

It is advisable in this design of pump to introduce the water on top rather than on the bottom. The location of the automatic relief valve allows the pump to free itself from air before the water enters the first impeller. The automatic operation of the discharge check valve can be accomplished in various ways, as shown in the illustrations.

FOOT VALVES.

So much trouble has arisen in practice because engineers have not appreciated the importance of a properly designed foot valve for a centrifugal pump that it has been deemed necessary to devote a separate section to this subject.

The construction of a centrifugal or volute pump is weak in itself, as the pumping head is formed at the periphery of impeller, and when the pump is working, the pressure on the side plates is much lower. These cannot be stayed in this form of pump, and are not supposed to stand the internal strain or pressure due to the total pump head. A sudden stoppage of the column of water traveling through the pipes causes a heavy pressure on the sides, tending to open and rupture the casing. The column of water is supported by the rotation of the impeller, and if from any cause this rotation suddenly ceases, the intensity of the reaction or shock is dependent on the weight of water in the pipes, and on the velocity acquired by the returning column before it is finally arrested. The sudden closing of the foot valve is frequently sufficient to split the pump casing, pipe and heads. This danger can be reduced by furnishing the foot valves with a relief or safety valve. This is particularly necessary when priming water is used under a pressure heavier than that for which the pump was built.

A relief valve of this kind can be made a part of the foot valve, or it can be attached to suction pipe and discharge back into the well. It is not, however, advisable to employ foot valves on large pumps, and other means for priming the pumps should be used.

Foot valves should have at least 150 per cent of the area of the suction

pipe, which should be the next size larger than discharge. Thus a pump with 10-inch discharge should have 12-inch suction pipe, and if a foot valve be employed its area should be one and one-half that of the 12-inch pipe to reduce the frictional resistance through the system; otherwise these losses

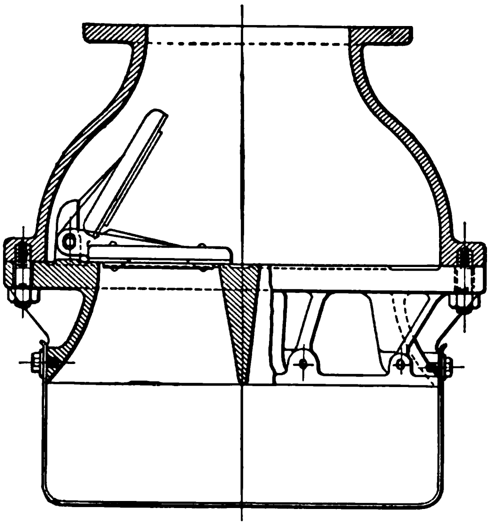


Fig. 5. Centrifugal Foot Valve.

absorb a large percentage of the total work in low-lift pumps, which means a poor economy, preventable with proper sizes of piping.

The valves should be of the flap design, so made that when open they rest on the sides of the body, thus allowing a clear passage for water through the center of the valves, as illustrated in Fig. 5.

CHAPTER V.

EFFICIENCY.

THE word efficiency in connection with centrifugal pumps has become very ambiguous, and has led to many disputes in connection with guarantees and contracts. This ambiguity has been brought about by the fact that the word has been used without modification to designate the efficiency of the pump only, of the pump and prime mover, and of the entire plant measured back to the boiler. This uncertainty can and should be eliminated. The efficiency of a centrifugal pump, when not otherwise modified, can mean but one thing, — the ratio of the water horse-power output at the pump to the brake horse-power input at the coupling or pulley.

The water horse-power output is determined by the total head against which the water is pumped and the quantity of water delivered. The usual method of finding the total head is to place a gauge on the suction line close to the pump, another on the discharge, and to note the vertical distance between the gauges. The algebraic difference between the gauge readings plus the vertical distance between the gauges, all in feet, is considered the total head. Another method is to add to the head thus obtained the velocity head in the discharge pipe. Still another method is to add to the difference in gauge readings and the vertical distance between gauges, the difference between the velocity heads in the suction and discharge pipes. It is therefore important in considering efficiencies that the total head be clearly interpreted.

The three methods of obtaining the total head may be represented by the following formulæ:

H = Total head in feet.
H_1 = Discharge head in feet.
H_2 = Suction head in feet.
A = Vertical distance between gauges in feet.
H_3 = Velocity head in discharge pipe in feet.
H_4 = Velocity head in suction pipe in feet.

First, $H = H_1 - H_2 + A$.
Second, $H = H_1 - H_2 + A + H_3$.
Third, $H = H_1 - H_2 + A + (H_3 - H_4)$.

H_3 and H_4 are made up of the flow in feet per second through the pipes thus:

$$H_3 = \frac{(\text{velocity in feet per second})^2}{2g},$$

in which g is the acceleration due to gravity, or 32.2.

Third formula represents the actual head pumped against and should always be used since the gauge reading shows the difference between the total head above the gauge and the velocity head. Where the suction and discharge pipes are of the same diameter, the velocity heads are equal, and equation (3) becomes the same as equation (1).

High efficiency in pumps is obtained by changing the kinetic energy of the water, as it issues from the wheel, into pressure, by reducing water friction, such as churning in the chambers, and the skin friction of rotating disks, and by reducing the friction of the bearings. The elements of total pump efficiency are therefore the absolute hydraulic efficiency, the mechanical efficiency, and the volumetric efficiency.

The absolute hydraulic efficiency η_{h+} expresses the ratio between the useful head and the total pumping head. If h represents the former and H the latter, then the total head H is made up of h and all frictional and shock losses of water in the pump, and is the most important factor to be considered. Information relating to these losses is very meager and incomplete. The most serious loss is due to skin friction between the impeller and the water. The head pumped against varies as the square of the velocity; hence the wasted power varies as the cube of the head. As the head increases, the loss from skin friction increases at a more rapid rate and thereby imposes a limit on the head against which the pump can be economically operated. The work wasted in disk friction varies as the square of the radius, hence a smaller impeller at a higher number of revolutions absorbs less power in friction than a larger one at a less number of revolutions but with the same peripheral velocity. The disk friction of the water near the axis is lower than that at the outer surface of the impeller. Experiments have been made abroad on the power lost by skin friction and the following formula has been obtained:

$$W = 8132 \times \frac{1}{n^2} \times h^{2.5} \text{ foot pounds.}$$

W = Power due to resistance in foot pounds,
n = Revolutions per minute,
h = Head in feet.

The largest losses by surface friction are along the walls of the casings, and were the surfaces of the stationary walls and impellers alike the water would receive a rotating motion one-half that of the impeller. These losses are reduced by having the internal walls smooth and impellers polished, and by proper clearance between impellers and casings. In addition to surface friction there are losses due to molecular friction in the water itself.

The mechanical efficiency depends upon the friction between the shaft, impeller, etc., and their bearings, and is a function of the workmanship, fit, and lubrication.

The volumetric efficiency is the ratio of the amount of water discharged to the amount entering the pump. The difference is the loss due to leakage in running fits. The greatest loss in a turbine pump is between the impeller and the diffusion ring. These losses vary from 2 to 10 per cent.

Efficiencies on large pumps have been obtained as high as 90 to 92 per cent, and in multistage turbines as high as 85 to 87 per cent. This has been accomplished by careful analysis, as shown above, by reducing the internal losses and skin friction, by dissipating shocks and disturbances and turning more of the velocity into effective head. The advance which this involves is shown by the fact that until recently the best efficiency obtained from the ordinary volute was 40 per cent. The conditions and approximate efficiencies which may be expected with correctly designed impellers are as follows:

Capacities from 75 gallons to 250 gallons will give about 55 to 65 per cent efficiencies;
250 gallons to 900 gallons, 70 per cent;
900 gallons to 3000 gallons, 70 to 73 per cent;
3000 gallons to 6000 gallons, 73 to 75 per cent; and
6000 gallons to 10,000 gallons, 75 to 78 per cent.
Above this we obtain from 75 to 85 per cent efficiencies.

A side entrance or single suction pump will give slightly less in each case. Sizes of the discharge pipes of pumps for the above vary from 4 to 60 inches diameter. Speeds for the small pumps vary from as high as 3500 revolutions per minute for the smallest down to 600 revolutions for 8- to 10-inch pumps. The larger may run as slow as 150 revolutions with good efficiency under favorable conditions.

The losses at the impeller outlets are due to the discharge velocity and can be expressed as $(aU^2\ 2\ g)$, where U is the absolute velocity of discharge from the vanes, and a is a coefficient with a value between 0.5 to 0.6.

The skin friction of the rotating impeller varies as the square of the peripheral speed, and also as the area of the casing. Considering first the loss of head on entering the impeller vanes, it must be assumed that the water enters the inner portion of the wheel radially. In order to avoid shock, the direction of the vane must coincide with the resultant of the radial velocity of the water and the tangential velocity of the inner circumference of the wheel.

The loss of head caused by changing of velocity at entrance of vane is

$$U^2\ 2\ g = V - V_2 \cot \theta.$$

The loss of head at outlet of impeller is

$$(V_3)^2\ 2\ g = V - V_3 (\cot \alpha + \cot \rho).$$

The remaining losses are due to friction and are proportional to the square

of velocity of flow through impeller and to the square of peripheral speed, and can be expressed as follows:

$$K(V_3)^2\,2\,g - L\,(V_1)^2\,2\,g,$$

K and L being constants.

The discharge head will decrease as the capacity is increased, the head depending upon V_1 and V_3, and is represented by the following formula:

$$(MV_1 - NV_3)^2\,2\,g,$$

where M and N are constants.

Therefore the total actual head equals the theoretical heads less the losses. The constants here mentioned should be determined from actual experiments of the particular type of pumps considered. A graphical illustration is shown in Fig. 6.

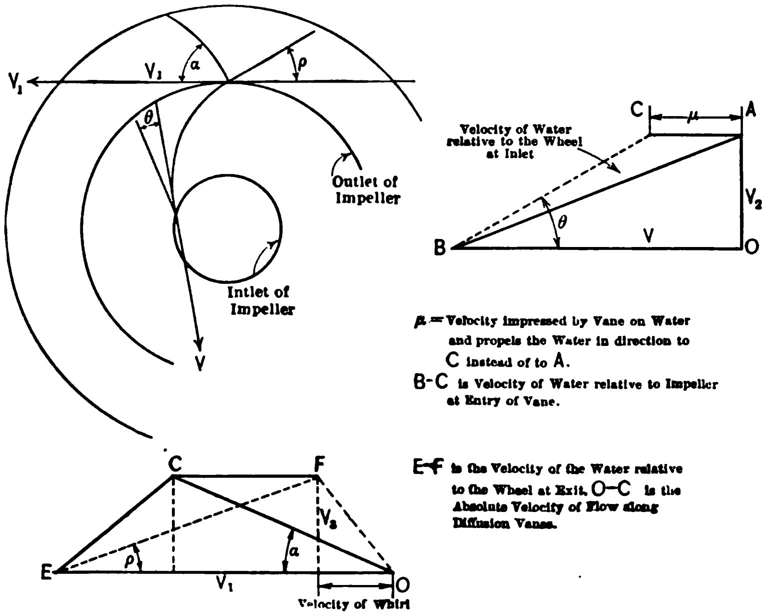
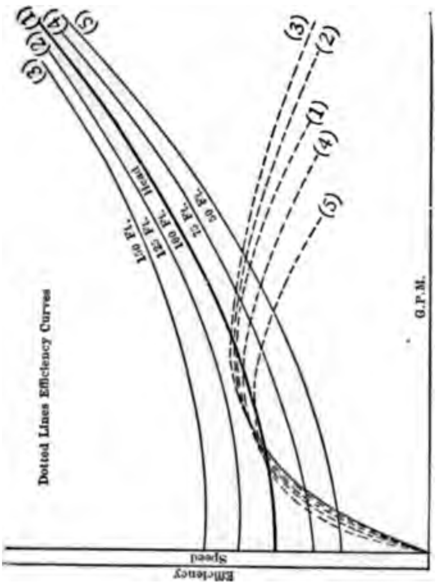


Fig. 6. Diagram Showing Loss of Head by Change of Velocity at Entrance of Vanes and at Outlet.



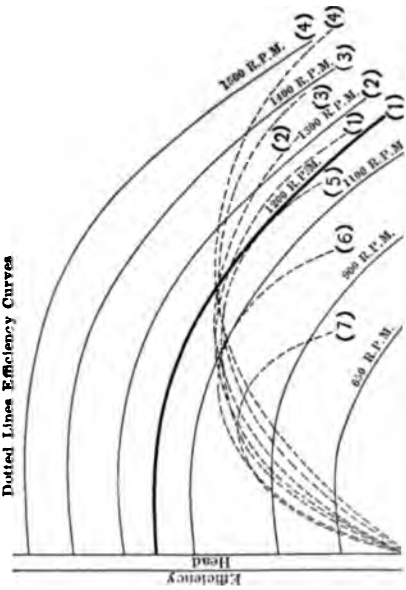
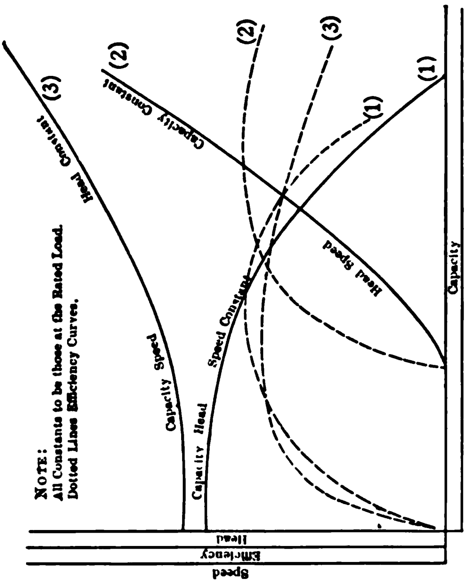


Fig. 9. Capacity and Speed for Constant Head.



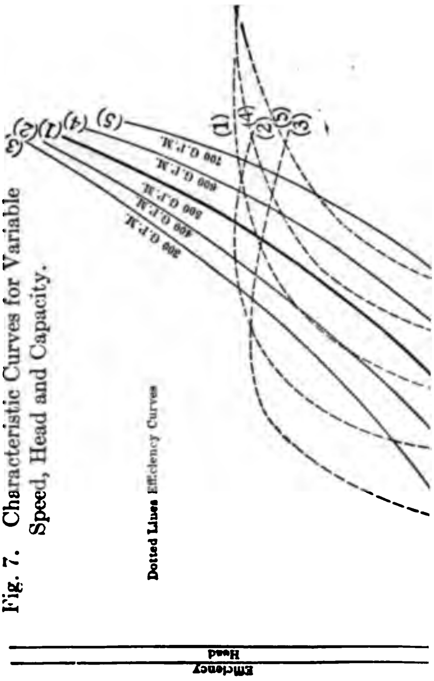
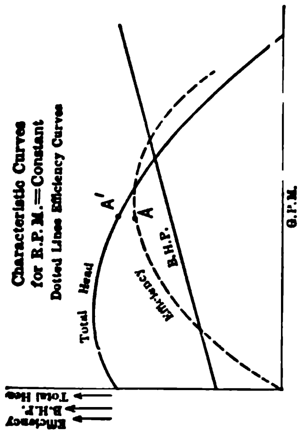


Fig. 7. Characteristic Curves for Variable Speed, Head and Capacity.



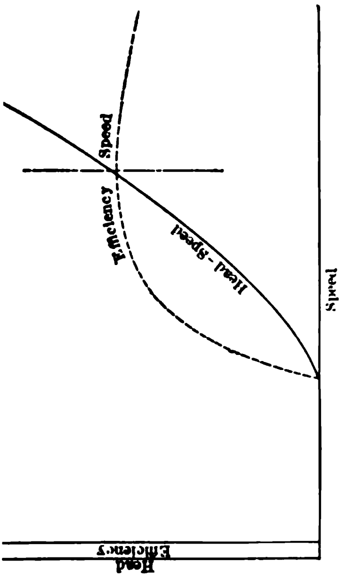


Fig. 11. Characteristic Curves for Volute Pump.

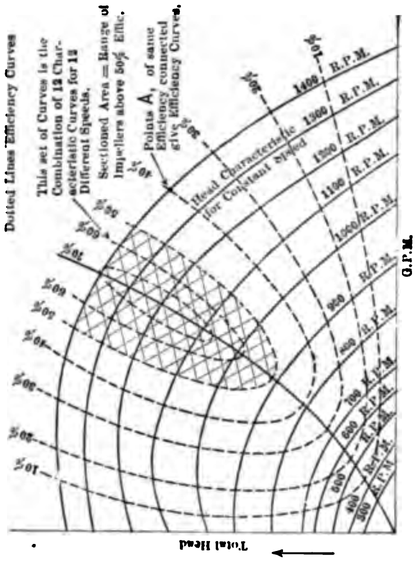


Fig. 12. Capacity-head Curves for Constant Speed with Interposed Constant Efficiency Curves.

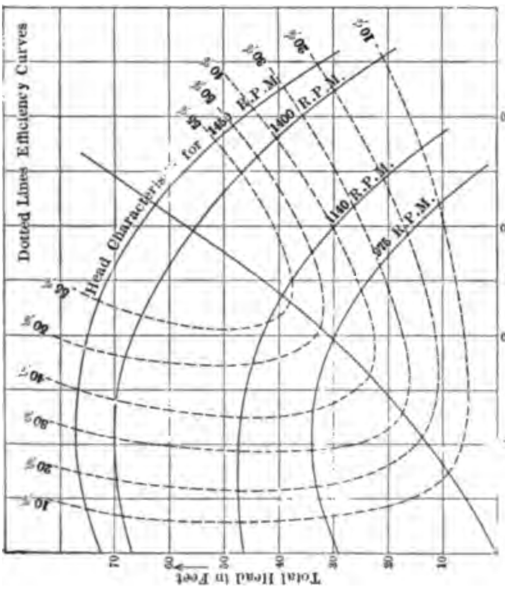


Fig. 13. Capacity-head Curves for Constant Speed with Interposed Constant Efficiency Curves.

CHAPTER VI.

CHARACTERISTICS.

Every impeller has a fixed relation between the head, capacity, and speed, known as the impeller characteristic. The prevailing idea that an impeller can be used with only one condition of head and capacity for a given speed is erroneous. An impeller is good for a certain range, and, in order to determine this, experiments must be made and curves plotted. These relations between capacity and head with the speed constant, head

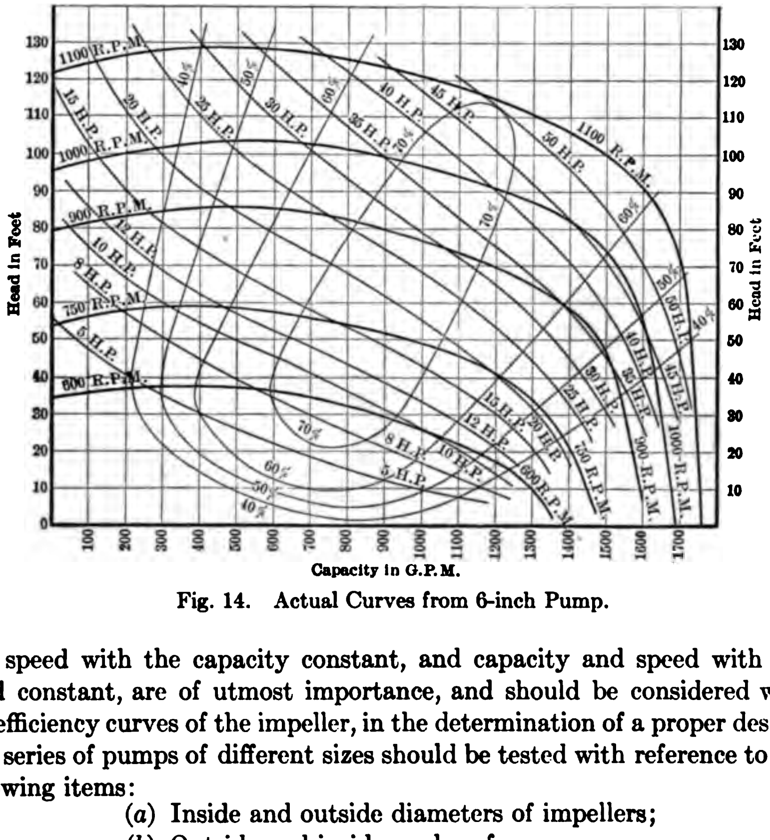


Fig. 14. Actual Curves from 6-inch Pump.

and speed with the capacity constant, and capacity and speed with the head constant, are of utmost importance, and should be considered with the efficiency curves of the impeller, in the determination of a proper design.

A series of pumps of different sizes should be tested with reference to the following items:

(*a*) Inside and outside diameters of impellers;
(*b*) Outside and inside angles of vanes;
(*c*) Radius and number of vanes;
(*d*) Width of impeller.

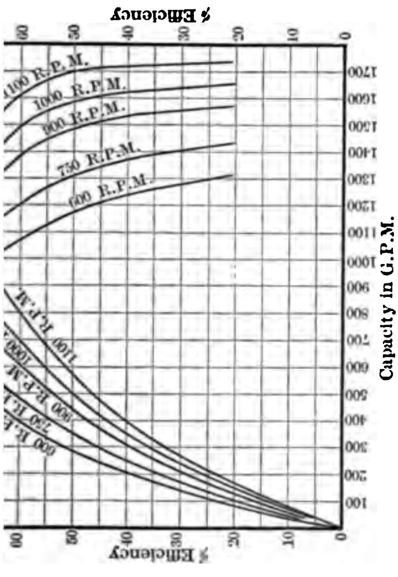


Fig. 15. Actual Curves for 6-inch Pump.

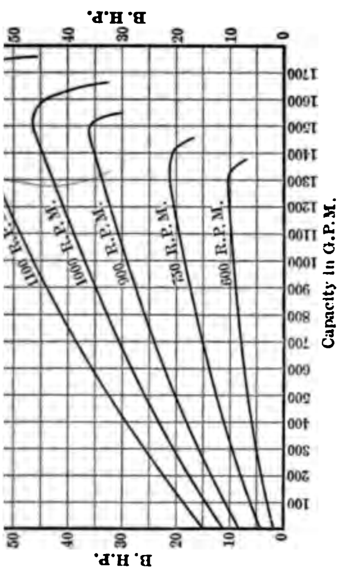


Fig. 16. Actual Curves for 6-inch Pump.

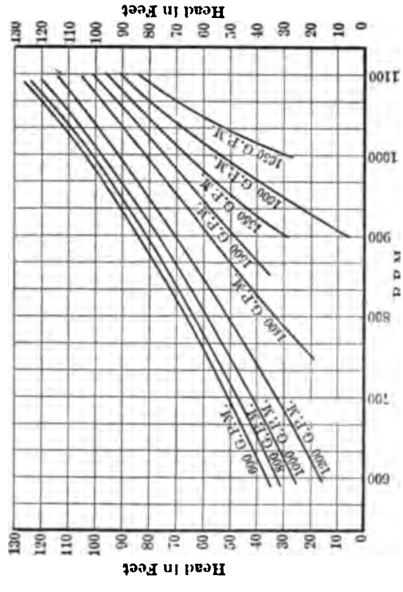


Fig. 17. Actual Curves for 6-inch Pump.

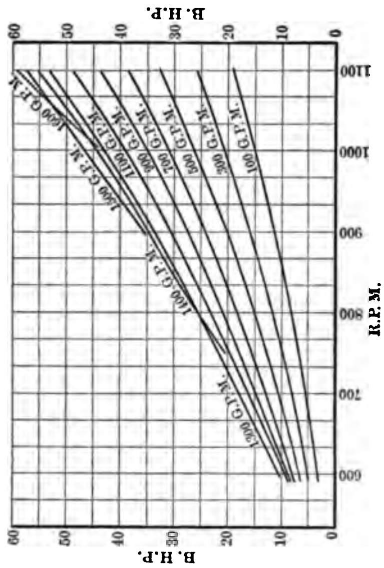


Fig. 18. Actual Curves for 6-inch Pump.

From these tests can be found that efficiency which will give with varying speed and constant capacity, the most advantageous peripheral speed for the given angles; and also that efficiency with varying capacity and constant speed, which will give the capacity most advantageous for a given width of impeller and inside and outside angles of vanes.

Fig. 7 shows a set of characteristic curves for these variables and constants with their respective efficiency curves in broken lines. Fig. 8 shows a set of speed and head curves with capacity constant for one im-

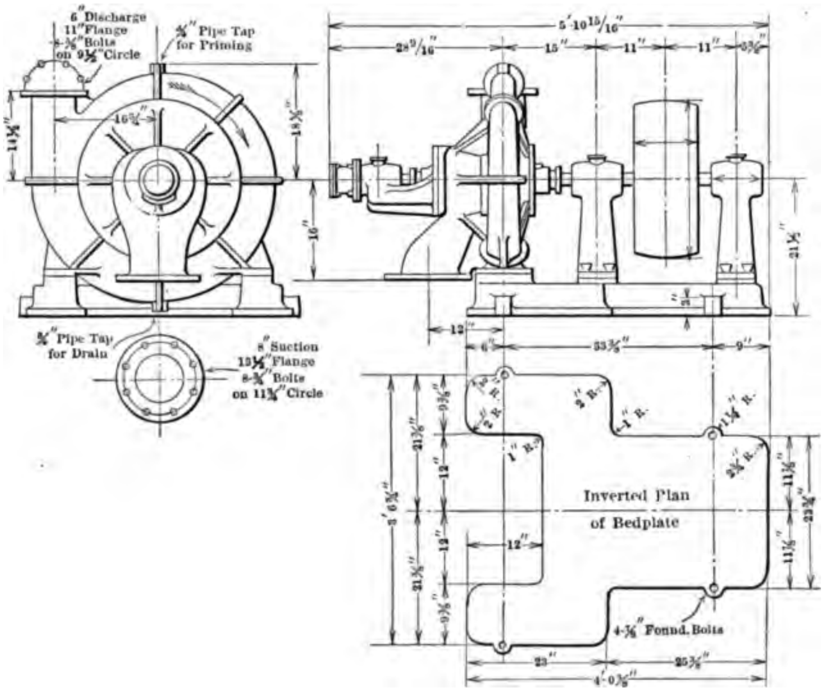


Fig. 19. Six-inch Pumps from which Actual Performance of Curves for Figs. 14 to 18 were taken.

peller. The dotted efficiency curves correspond to the speed-head curves bearing the same numbers. Fig. 9 shows a set of capacity-speed curves for constant head. Fig. 10 shows a set of capacity-head curves for constant speed. Fig. 11 gives characteristics for a regular volute pump which can be compared with turbine or diffusion pumps.

Figs. 12 and 13 show a series of capacity-head curves for constant speed upon which have been interposed a series of constant-efficiency curves. The method of plotting these efficiency curves is shown at the top of Fig. 12. From the capacity-efficiency curve, a given efficiency A is projected to the corresponding capacity ordinate on the capacity-head curve at A,

and points of equal efficiency for the various speed characteristics are then connected. A series of these equal-efficiency curves will show the maxi-

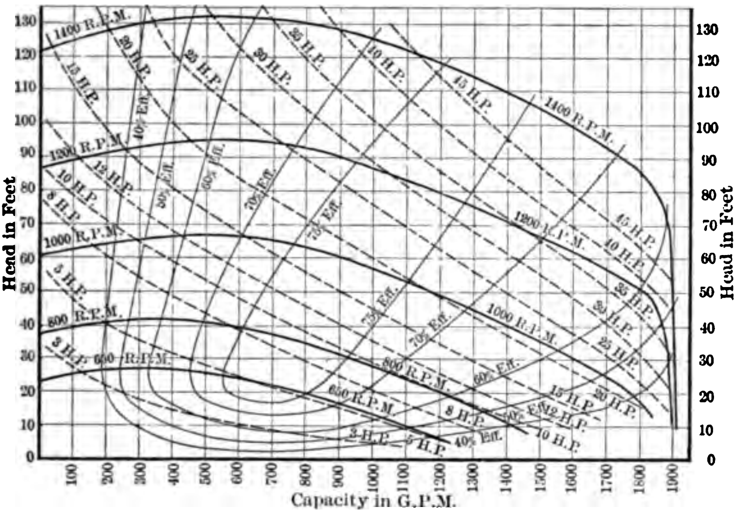


Fig. 20. Actual Curves from a Special Designed Pump.

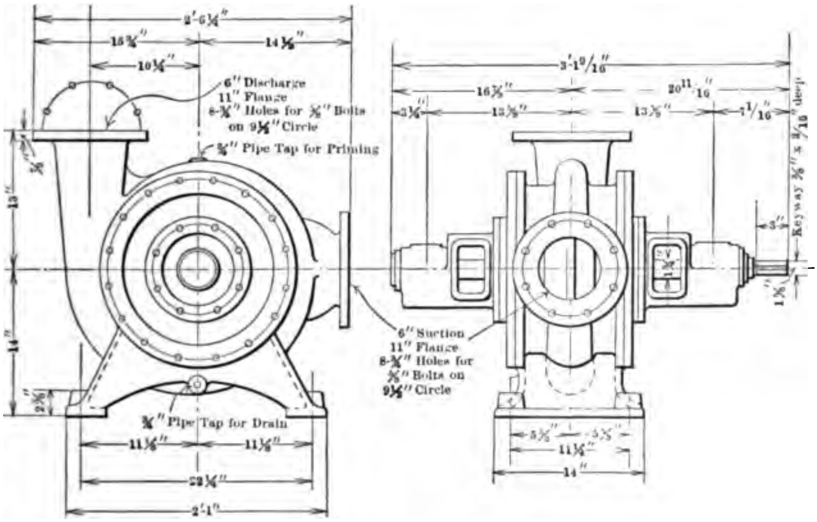


Fig. 25. Special Designed Pump from which Curves for Figs. 20 to 24 were taken.

mum efficiency for any speed with a given impeller, and is very useful, since the usual condition is constant speed.

For constant speed the relations usually required are those existing between capacity and head, capacity and power, and capacity and efficiency.

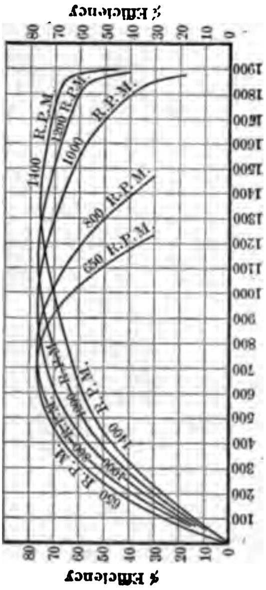


Fig. 21. Actual Curves for a 6-inch Special Design Pump.

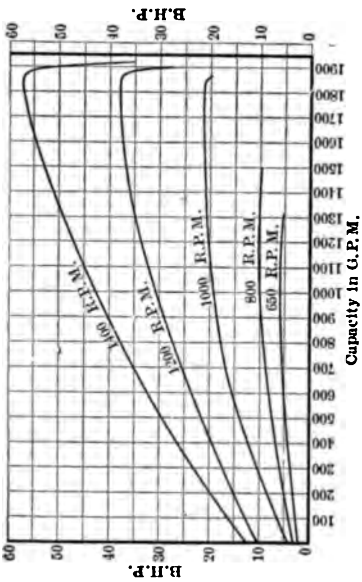


Fig. 22. Actual Curves for a 6-inch Special Design Pump.

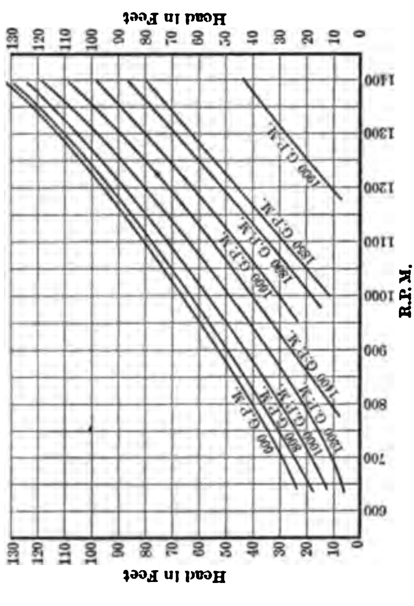


Fig. 23. Actual Curves for a 6-inch Special Design Pump.

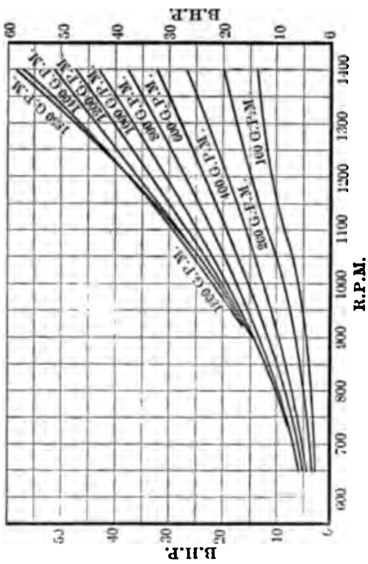


Fig. 24. Actual Curves for a 6-inch Special Design Pump.

Figs. 14 to 18 inclusive show curves taken from actual tests of a 6-inch volute type of pump illustrated in Fig. 19. Figs. 20 to 24 show curves taken from a 6-inch special type of pump illustrated in Fig. 25.

These illustrations show fully the characteristics of these pumps and illustrate the possibilities of a careful analysis of the subject, and point out the way to reach results by the simplest and most reliable road.

The characteristics show advantages of centrifugal pumps not possessed by any other type of pump. It is impossible to produce a pressure in the pipe lines higher than that corresponding to the speed as shown from the form of the curve, obviating all danger of line breaks when a valve on the discharge line is accidentally closed. The capacity-horse-power curves show that a sudden closing of the discharge valve will reduce the load on the motor. On the other hand, the usual types of centrifugal pumps overload the motors should a break occur in pipe line reducing the pump head and increasing the capacity, and in order to care for this the impellers should always be designed so that not more than 25 per cent overload can be thrown on the motor.

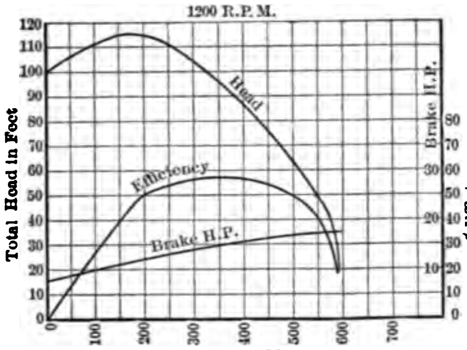


Fig. 26. Characteristic Curve for Fixed Overload Condition.

It is to be noted that when the head pumped against is reduced an increase in the discharge takes place, due to the fact that the surplus head produced by the pump is converted into a velocity head. Increasing the speed may also overload the motor, but the impeller can be so designed that 25 per cent can be made the maximum overload. The characteristics of such pumps are shown in Figs. 26 and 27, from which it can be seen that the maximum power at constant speed for greatest variation in head is only 1.25 times the power absorbed by the pump when giving the highest efficiency.

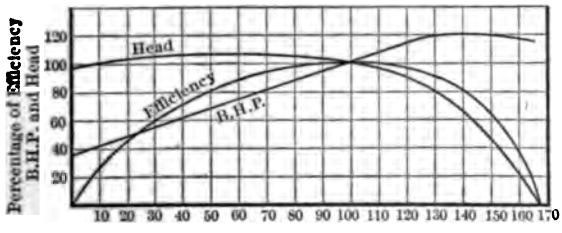


Fig. 27. Characteristic Curve for Fixed Overload Condition.

A pump can be so designed that the power curve will be nearly constant

while the efficiency is maintained. The ordinary design, however, gives a capacity varying from 70 to 125 per cent, with a variation of head of 25 per cent, and an efficiency variation of about 5 per cent.

In Figs. 55 and 58 the various forms of impeller vanes show the different characteristics as usually applied to commercial problems. No. 3 will maintain with constant speed a uniform efficiency under a varying head. No. 2 will produce a constant head for the best efficiency, but has the disadvantage that the power increases rapidly. No. 1 would overload considerably and is limited to but few conditions, as when the load is variable, due to the frictional resistances other than the regular pumping head. The extent to which a designer can go in the development of the shape of the vanes depends entirely upon the conditions of service, and should be carefully investigated.

CHAPTER VII.

OPERATING.

Much trouble may be experienced in operating centrifugal pumps if due consideration is not given to the physical conditions of the pipe line and installation and to the material to be pumped. The following paragraphs should be carefully considered before erecting and starting a centrifugal installation.

Suitable foundations are essential for satisfactory and continuous operation. The foundation may consist of anything, so long as it is homogeneous and stiff enough to support the bedplate at all points when the pump is in operation. Without proper supports, bedplates cannot be expected to maintain the proper alignment of the pump shaft.

Erection. — A combined bedplate for pump and motor should be leveled up by wedges, the pump and motor placed upon the facing strips and lined up so that the faces of the pump coupling are parallel, and the pump and motor run freely with and without coupling bolts in position. The bedplate should then be grouted into place so that it is absolutely rigid. After the foundation bolts have been permanently set the suction and discharge piping may be connected.

Suction Piping. — Care should be taken to see that there are no air pockets in the suction piping and that all joints are absolutely air-tight. The pump should be placed as near as possible to the level of the water in the suction well, but in cases where the supply is distant there should be a continuous rise in the suction pipe toward the pump.

The suction lift or vertical distance from top of impeller to suction water level should never exceed 25 feet at sea level, 20 feet at ½ mile above sea level, and 16 feet at 1 mile above sea level.

Discharge Piping. — The discharge piping should be as direct as possible and with a minimum number of bends. A tapered section should be placed at the pump discharge to increase the diameter of the delivery to the same size as the suction. If a centrifugal pump has too much resistance, the whole power of the pump may be expended in overcoming the frictional head, and give no discharge, hence the necessity of giving the water favorable passage during its travel through the pipes and the pump and avoiding all unnecessary resistance. The total head, theoretically, which a centrifugal pump will produce is $\frac{V^2}{2g}$, where V is the peripheral velocity of the impeller, and the discharge valve is entirely closed. The peripheral

velocity required to support a column of water of given height is equal to the velocity acquired by a body when it has fallen through a height equal to that of the column. This velocity is usually expressed

$$V = \sqrt{2\,gH} \text{ or } \sqrt{64.4 \times H},$$

where H is the total vertical distance in feet from the suction level to that of the discharge. All frictional resistances must be included in this. To this must also be given an increment in order to effect a discharge or delivery of water. In carefully designed and highly efficient pumps, the periphery velocity is found to be less than that given above. The particular equations later described in this book, were developed on the basis of a complicated theory obtained from the characteristics of each type and size of pump under close tests. When the water has been set in motion the pump will continue to deliver even when the velocity only equals $\sqrt{2\,gH}$.

Lubrication of Bearings. — After erection and before starting, all pump bearings should be carefully washed out with clean petroleum. It is most essential that the oil used for lubrication should be of the best quality for high speeds, and absolutely free from all trace of grit or sediment.

Priming. — It is essential to have a gate valve at the point of discharge from the pump, which should be closed before the pump is started. The pump should then be primed in one of the ways described in Chapter IV. When the pump and suction pipe are filled and all air expelled, the air cocks should be shut and the various pump and motor bearings carefully examined before the motor switch is closed. Pumps should never be run without water.

Starting. — As a rule, the pump will come up to normal speed in about twenty seconds, after which the gate valve on the discharge should be very slowly opened, allowing the full load to come on the motor as gradually as possible.

Stuffing Boxes. — The packing in the pump stuffing boxes should be carefully adjusted and on no account should the glands be screwed too tight. To prevent air leakage through the stuffing box on the suction side of the pump, a gland cage may be provided within the stuffing box. On each side of the cage there should be placed about three rings of good hydraulic graphite packing. A ¼-inch pipe from the discharge of the pump should be run to the stuffing box and an opening connected through to the cage. This makes a water seal in the stuffing box and prevents all air leakage. The gland should be run as loose as possible, otherwise the packing is liable to cut the shaft. A small amount of leakage from the stuffing box does no harm, in fact is an advantage, as it prevents the packing from heating and at the same time keeps it lubricated.

Suction Foot Valve and Strainer. — In all centrifugal pumps where there is a suction lift, it is absolutely essential that a foot valve of proper design

should be fitted to the suction pipe, unless special means are provided for emptying the pump and suction pipe. Careful attention should be given to this valve, which is described in Chapter IV. The foot valve should in all cases be fitted with a strainer having an area of not less than twice the cross-sectional area of the pipe.

Before starting the first time it is advisable to clean out thoroughly the bearings, including the thrust bearing, by pouring in kerosene and allowing it to run out at the bottom, as dirt is liable to get into the bearings during shipment. The bearing should then be filled as full as possible with a first-class lubricating oil similar to dynamo oil. After priming, the pump can be started and brought up to speed with the discharge valve closed. The discharge valve can then be opened until the desired quantity of water is obtained. If the total head operated against is greater than that for which the pump was designed, the quantity of water discharged will be less than full capacity, and there may even be no discharge at all if the head is sufficiently great, but the power will be less than that required by the designed head. If the total head is less than the head for which the pump is designed, the amount of water discharged will be greater than the normal capacity of the pump, and the power consumed will also be greater than the normal rating.

The pumps can be run on a reduced head by throttling the discharge until the desired quantity of water is obtained. This creates a friction head and reduces the efficiency of pumping, but will prevent overloading the motor.

In operation, the only attention which the pumps require is an occasional inspection of the oil in the bearings and of the packing in the stuffing boxes.

PART II.

CHAPTER VIII.

GENERAL REMARKS.

DISCUSSION OF THE CENTRIFUGAL PUMP.

No theory of the centrifugal pump has been written. In making this statement the word "theory" has been used in its correct sense, which means the formulating of the results of observation into laws and not mere hypothesis. A clearer idea of the significance of the word "theory" would cause less dispute and show the fallacy of the common expression, "It may be right in theory but wrong in practice." This is a contradiction, since the theory cannot be correct if it does not fully agree with practice. Hypothesis is really meant, and the statement should really be, "The hypothesis or assumption does not agree with facts." Men have been criticized for being theoretical, and fault has been found with their mathematics when the real trouble lay in a misconception of the theory of the problem and a consequent incorrect application of mathematics.

The so-called theories of centrifugal pumps are analyses based on certain assumptions. Three so-called theories will be presented here which will give approximately the same results. The basis of the principal one of these is that the motion of the particles of water throughout the space between the blades strictly follows those particles contiguous to the blades. Another assumes that the water enters the inner periphery of the wheel in a radial direction across the entire width of the wheel. Both of these are wrong and are made only because the problem would otherwise present too many difficulties, and in fact be impossible of solution. There are several analyses of the theory of centrifugal pumps, intricate but very useful to the student, and authors have discussed the problem in a masterly manner. Such writers as the Russian engineer, T. H. Brix, Dr. Egon R. v. Greenebaum, and Engr. Fritz Neuman have given complete analyses. Unfortunately, there is no such analysis in English. The formulæ given in this book will be such as will be readily available for the designer.

Wheels usually have their blades made in the form of an arc or arcs of circles. An exception is sometimes found in commercial standard wheels where the outer extremity has the form of a cycloid, drawn by eye. Some wheels have blades of the form of an Archimedian spiral.

CHAPTER IX.

FIRST THEORY OR ANALYSIS.

THEORY OF IMPELLERS.

THE power required to rotate the wheel is based upon the change in the velocity of the water as it passes from the inner to the outer periphery of the wheel and is expressed by the equation

$$\frac{W}{g}(VU - vu) = \frac{WH}{E},$$

in which

W = the weight in pounds of the water pumped per second;
H = the vertical distance in feet between the levels of the water in the suction and discharge reservoirs;
v and V = the tangential velocities of the wheel in feet per second at its inner and outer periphery respectively;
u and U = the tangential velocities in feet per second of the water at the inner and outer periphery of the wheel respectively;
g = the acceleration due to gravity;
E = the hydraulic efficiency of the wheel.

The left-hand member of the above equation represents the work required to overcome the hydraulic resistance of the wheel, but does not include any mechanical resistances such as friction of the bearings, etc. The right-hand member represents the work done.

This equation contains the fundamental principle used in calculating ordinary centrifugal pumps as given by most of the writers. As stated, it is assumed in ordinary calculations for centrifugal pumps that the water enters the eye of the wheel radially, and consequently the term vu becomes zero, and we may write $V = \frac{gH}{EU}$. From the triangle of velocities in Fig. 28 it will be easy to deduce the following formulæ:

$$V = \sqrt{\frac{gH}{E}\left(\frac{\sin(A-B)}{\sin A \cos B}\right)} = \sqrt{\frac{gH}{E}\left(1 - \frac{\operatorname{tang} B}{\operatorname{tang} A}\right)} = \sqrt{\frac{gH}{E}\left(1 + \frac{\operatorname{tang} B}{\operatorname{tang} D}\right)}.$$

$$V = \sqrt{\frac{gH}{E}\left(1 + \frac{\cot T}{\cot C}\right)},$$

$$\frac{\operatorname{tang} B}{\operatorname{tang} A} = \frac{\operatorname{tang} B}{\operatorname{tang} D} = \frac{\cot A}{\cot B} = -\frac{\cot T}{\cot C} = 1 - \frac{EV^2}{SH},$$

and
$$\frac{V}{J} = \cot B - \cot A = \tan T + \tan C,$$

$$\cot D = \tan C = \frac{V}{J} - \frac{gH}{EJV},$$

$$\cot B = \frac{gH}{EJV}.$$

The above embrace all the formulæ given in various forms. The greater the number of blades the more nearly will the assumption agree with the facts, but too many blades will cause undue friction. Brix and other writers give complicated formulæ for the number of blades, but this is usually

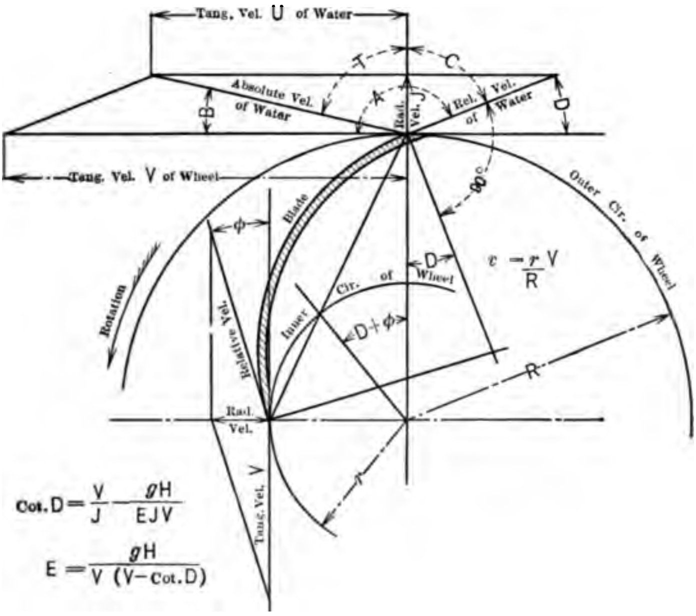


Fig. 28. Triangle of Velocities.

assumed. The diameter, speed, and width of the wheel are dependent on the motor and on the allowable velocity of the wheel and the water through it, features which each designer will have to settle.

The angle of the blade as well as its radius is determined by the following geometrical and analytical considerations. To construct the curve of the blade, it is necessary to know the inner and outer radii of the wheel, and the angles ϕ and D which the blade makes with the inner and outer periphery of the wheel. The angle D is determined by calculation from formulæ given, and usually the radial velocity at the inner periphery of the wheel is assumed to be the same as that at the outer; the angle ϕ, unit of blade, may be graphically determined as shown in Fig. 29.

Referring to Fig. 29 we proceed as follows: Let r and R be the radii of the wheel. From a draw a line, making the angle D as shown, and a line from the center O, making the angle $D + \phi$. From a draw the line abc; then c will be the inner extremity of the blade. Draw the line de perpendicular to and bisecting the line ac; then the point d will be the center of the curve of the blade as shown. This assumes that the angles ϕ and D are known.

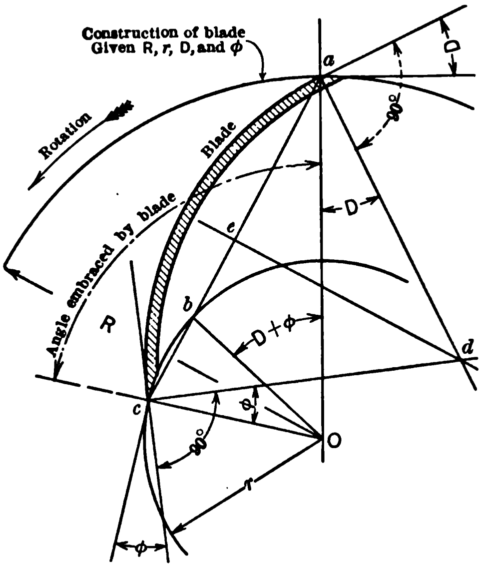


Fig. 29. Construction of Curve of Blade.

To find the angle ϕ, the velocity v at the inner circumference of the wheel varies inversely with ratio of the radii and directly with the velocity of the wheel at the outer circumference, that is,

$$v = \frac{r}{R} \times V.$$

Since it has been assumed that the water enters the eye of the wheel radially, it has no velocity of rotation, and the triangle of velocities is as shown in Fig. 30.

If Q be the volume of water in cubic feet pumped per second, w the width of the wheel at the outer circumference, then, disregarding the thickness of the blades, the radial velocity of the water at the outer circumference of the wheel will be

$$\frac{Q}{2\,\pi r w},$$

all dimensions being in feet.

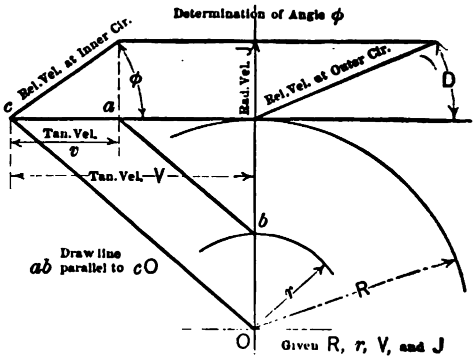


Fig. 30. Triangle of Velocity with Water Entering Wheel Radially.

The relative velocity of the water at entrance and exit of the wheel should have the same direction as that of the blade at the corresponding points, and this is what governs the angles D and ϕ. If this is not followed injurious shocks will result.

This discussion does not consider the influence of a vortex chamber or the form of the discharge passage around the wheel.

The calculation of an 8-inch three-stage pump, with details of impeller and diffusion vanes, according to the above analysis, is shown in Fig. 31 and described below.

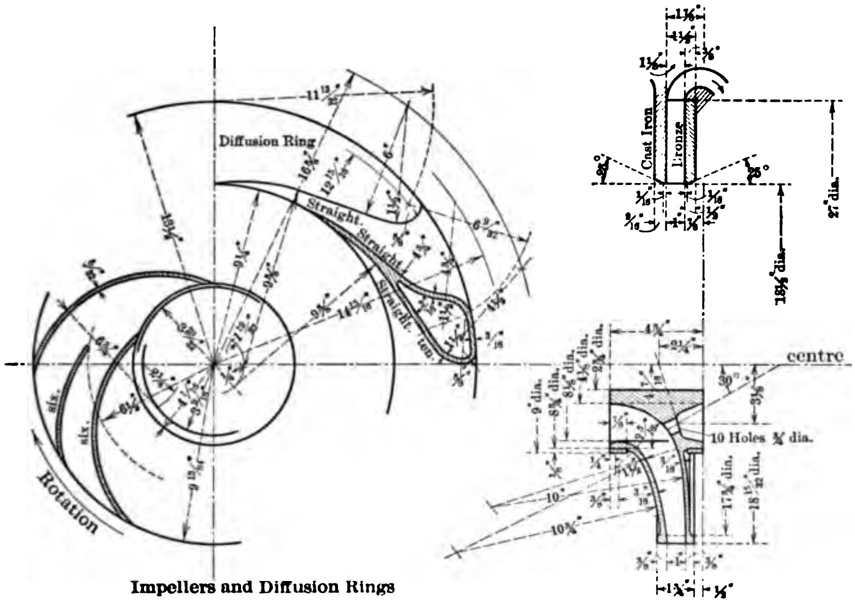
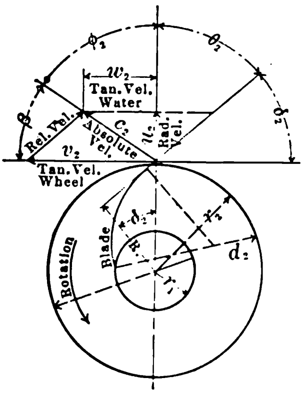


Fig. 31. Impeller and Diffusion Vanes of 8-inch 3-stage.

CALCULATION OF 8-INCH 3-STAGE

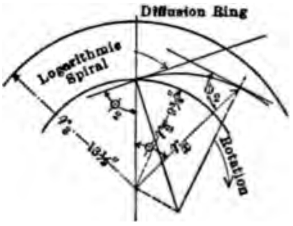
Conditions: 1400 gallons per minute.
1170 revolutions per minute.
210 pounds pressure or 70 pounds, each stage.

Outer diameter of impeller	18 15/32″
Inner diameter of impeller	8 1/8″
Hub diameter of impeller	4 1/8″
Width of opening at outer circumference of impeller	1″
Width of opening at inner circumference of impeller	2 3/4″
Number of blades	12
Thickness of blades at outer circumference	3/8″
Outer diameter of diffusion ring	27″
Inner diameter of diffusion ring	18 1/2″
Number of guides in diffusion ring	10



Formulæ Used.

$$v_2 = \sqrt{\frac{gH}{\eta}\left(1+\frac{\tan\theta_2}{\tan\phi_2}\right)} \qquad \frac{v_2}{u_2} = \tan\phi_2 + \tan\theta_2.$$



$$\eta = \frac{gH}{v_2^2}\left(1+\frac{\tan\theta_2}{\tan\phi_2}\right) \text{ efficiency.}$$

$$= \frac{gH}{v_2u_2}\,\frac{1}{\tan\phi_2} = \frac{gH\cot\phi_2}{v_2u_2}.$$

H = net or useful, here 70 pounds head.
All dimensions in feet; velocity in feet per second.

$$r_x r_2 e^{\cot\phi_2 \times \frac{\phi^\circ\pi}{180}}.$$

$$\log r_x = \log r_2 + \cot\phi_2 \times \frac{\phi^\circ\pi}{180} \times \log e.$$

For equal steps of ϕ° there is a constant difference, for $\log r_x = \left\{\frac{\pi\cot\phi_2}{180}\log e\right\}\{\phi_{11}{}^\circ - \phi_1{}^\circ\}$.
These dimensions in inches.

$$v_2 = \frac{18.46875 \times 1170 \times \pi}{12 \times 60}.$$

d_2 – 18.46875 log 1.2664374
π log 0.4971499
1170 log 3.0681859
4.8317732
$60 \times 12 = 720$ log 2.8573325
$v_2 = 94.284$ ft. log 1.9744407

18.46875 log 1.2664374
π 0.4971499
1.7635873

Outer circumference of impeller = 58.02 in.
Blades $12 \times \frac{3}{8}$ = 4.50 in.
Net outer circumference 53.52 in.

As opening is 1 inch wide this is also the *area* of the opening in square inches.

$$u_2 = \frac{\frac{1400 \times 231}{60}}{12 \times 53.52} = \frac{5390}{12 \times 53.52} = 8.3925 \text{ feet per second.}$$

5390 log 3.7315888
53.52 log 1.7285161
12.00 log 1.0791812
2.8076973

8.3925 log 0.9238915

$R = 8\frac{21}{32}$
$r_2 = 9\frac{15}{64}$
$\cos\delta_2 = 8\frac{21}{32} \div 9\frac{15}{64} = \frac{554}{591}$

log 554 2.7435098
591 2.7715875
log cos of 20°–22′–53.55″ = $\bar{9}.9719223$

$\cos\delta_2 = \sin\theta_2$ $\tan\theta_2 = \cot\delta_2.$
δ_2 = 20°–20′–53.55″ log cot 0.4307806
or tan of θ_2 = tan 69°–39′–6.45″

v_2 = 94.284 log 1.9744407
u_2 8.3925 log 0.9238915
$\frac{v_2}{u_2}$ 11.234 log 1.0505492

tan nat. θ_2 2.692
tan nat. ϕ_2 8.542

$\phi_2 = 830\text{-}19'\text{-}22.28''$. $\tan\theta_2 = 2.692$ log 0.4307806

$\tan\phi_2 = 8.542$ log $\underline{0.9315596}$

$\tan\theta_2 \div \tan\phi_2 = 0.31566$ log 9.4992210

$g = 32.16$ log 1.5073160

$H = 70$ lbs. $= 161.63$ ft. log $\underline{2.2084414}$

3.7157574

v_2^2 log $\underline{3.9488814}$

9.7668760–10

$\left\{1 + \dfrac{\tan\theta_2}{\tan\phi_2}\right\} = 1.31566$ log $\underline{0.1191007\text{–}10}$

Efficiency $\eta = 0.76908$ log 9.8859767

This is the hydraulic efficiency as computed from the formula.

Diffusion Ring.

Calculation of one radius vector of the curve of guide.

$$\log r_x = \log r_2 + \cot\phi_2 \times \frac{\phi^\circ \times \pi}{180} \times \log e.$$

Take $\phi^\circ = 36^\circ$; then

$$\log r_x = \log 9.25 + \cot\phi_2 \times \frac{\pi}{5} \times \log e$$

$\cot\phi_2$ log 9.0684404–10

π log 0.4971499

$\log e$ log $\underline{9.6377843\text{–}10}$

9.2033696–10

5 log $\underline{0.6989700}$

0.0319447 log 8.5043996–10

log 9.25 0.9661417

$\frac{\pi}{5}\cot\phi_2 \log e$ $\underline{0.0319447}$

0.9980864

$r_x = 9.956''$

$r_2 = \underline{9.25''}$

diff. $= 0.706''$

Gain in pressure in free vortex $= \dfrac{\omega_2^2}{2g}\left\{1 - \left(\dfrac{r_2}{r_3}\right)^2\right\}$

$\omega_2 = v_2 - \dfrac{u_2}{\tan\delta_2}$ $u_2 = 8.3925$ log 0.9238915

$\tan\delta_2$ $\underline{9.5692194\text{–}10}$

22.629 9.3546721–10

$v_2 = 94.284$

$u_2 \div \tan\delta_2 = \underline{22.629}$

71.655

$v_2 \dfrac{u_2}{\tan\delta_2}$ log 71.655 or $\omega_2 = 1.8552464$

$\log\omega_2^2 = \underline{3.7104928}$

$$\left\{1 - \left(\frac{r_2}{r_3}\right)^2\right\} = 1 - \left(\frac{37}{54}\right)^2 = \frac{1547}{2916}$$

1547 log 3.1894903

2916 log $\underline{3.4647875}$

9.7247028–10

$2.9 = 64.32$ log $\underline{1.8083460}$

0.0082481 log 7.9163568–10

ω_2^2 3.7104928

$1 - \left(\dfrac{r_2}{r_3}\right)^2$ $\underline{7.9163568\text{–}10}$

1.6268496

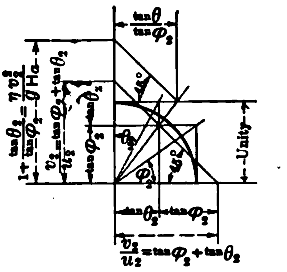
42.349 ft. gain in head, or 18.34 pounds.

To set this before the reader in simple form, the following diagrams, with calculation, have been developed, which can be applied to impellers and diffusion vanes in turbine pumps (Fig. 32).

STUDY OF CENTRIFUGAL PUMP.

Radial velocity = cubic feet per second divided by the net circumferential area in square feet; that is, allowance must be made for thickness of blades.

Fig. 32. Diagram for Calculating Impellers and Diffusion Vanes.



Graphical Solution.

Ha = useful lift in feet.

$\eta < 1$ = efficiency.

All dimensions in feet.

All velocities in feet per second.

$$\eta = \frac{gHa}{v_2 w_2}.$$

$$\left(\frac{gHa}{\eta}\right)^2 = v_2^2 w_2^2$$

$$= v_2^2(c_2^2 - u_2^2)$$

$$= v_2^2 c_2^2(1 - \sin^2 \beta_2)$$

$$= v_2^2 c_2^2 \cos^2 \beta_2$$

$$= v_2^4 \frac{\sin^2 \alpha_2 \cos^2 \beta_2}{\sin^2 (\alpha_2 - \beta_2)};$$

hence,

$$v_2 = \sqrt{\frac{gHa}{\eta}\left(1 - \frac{\tan \beta_2}{\tan \alpha_2}\right)},$$

$$\frac{\eta v_2^2}{gHa} = 1 - \frac{\tan \beta_2}{\tan \alpha_2} = 1 + \frac{\tan \theta_2}{\tan \phi_2}. \qquad (1)$$

$$c_2 = \frac{u_2}{\cos \phi_2}, \qquad z_2 = \frac{u_2}{\cos \theta_2},$$

$$v_2 = c_2 \sin \phi_2 + z_2 \sin \theta_2$$

$$= u_2 \frac{\sin \phi_2}{\cos \phi_2} + u_2 \frac{\sin \theta_2}{\cos \theta_2}$$

$$= u_2 \tan \phi_2 + u_2 \tan \theta_2,$$

$$\frac{v_2}{u_2} = \tan \phi_2 + \tan \theta_2. \qquad (2)$$

Curve of Guides in Whirlpool Chamber.

$r_x = r_2 e^{\phi \cot \phi_2}$ (ϕ in circular measure)

or $\log r_x = \log r_2 + \frac{\phi^\circ \times \pi}{180} \times \cot \phi_2 \log_e.$

Or we may calculate one value, such as r_4, and construct all others. The vector bisecting the angle between any two vectors is a mean proportional between these two vectors. Thus, $r_3^2 = r_2 r_3$, a property of the logarithmic spiral. Describe semicircle *bac*, erect perpendicular *Oa*, draw arc *ad* from *O* as center, and *d* gives point in curve. Continued bisections will give new points.

To draw the blades: Lay off angle $\delta_1 + \delta_2$, as shown, and line through *f* gives *h*, the inner end of blade. Draw line δ at an angle δ_2, as shown, and the point *p* at the intersection of line perpendicular to and bisecting line *hf* is the center of the curve of blade.

THEORY OF DIFFUSION GUIDES AND VANES.

The calculation of the curve of the diffusor guides or vanes in the chamber is shown in Fig. 33. Let r be the distance from the center of the wheel to any point of the curve, and B the angle which the tangent to the curve at that point makes with radius, which is constant at all points, this being a property of the logarithmic spiral. This angle B is also that which the absolute velocity of the water as it leaves the wheel makes with the tangent to the wheel at the outer circumference. We then have as the equation of the path of the water in the chamber,

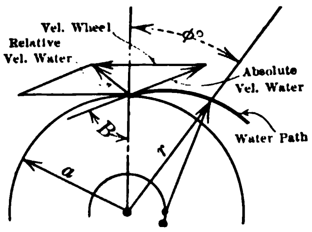


Fig. 33. Velocity Diagram for Diffusion Vanes.

$$\log r = \log a + \cot B \times \phi \times \log e,$$

where a is the radius of the outside of the wheel, ϕ is the angle in *circular measure* which the radius through the point in the curve makes with the radius of the point at the outer circumference of the wheel, and e is the Napierian base.

If $\phi°$ is the angle in degrees, then $\phi = \phi° \times 3.1416 \div 180$.

Example. —

$$a = 9.25 \text{ inches};$$
$$B = 80°;$$
$$\phi = n \times 10° \times 3.1416 \div 180,$$

where n is a multiple of an angle of 10°. We then have

$$\log r = \log 9.25 + (\cot 80 \times n \times 10 \times 3.1416 \times \log e \div 180).$$

cot 80	log 9.2463188 − 10
cot 10	log 1.0000000
3.1416	log 0.4971499
log e	log 9.6377843 − 10
	0.3812530
180	log 2.2552725
0.0133654	log 8.1259805 − 10
9.25	log 0.9661417

$$\phi° = \frac{(\log r - \log a) \times 180}{3.1416 \times \cot B \times \log e}$$

$\log r = 0.9661417 + 0.0133654 \times n$.
If $\phi° = 30°$, then $n = 3$ and
$\log r = 0.9661417 + 0.0400962$
$= 1.0062379$ and $r = 10.14467$.

Suppose r is given and we wish to find the corresponding angle $\phi°$, then we have

$r = 10.14467$	log 1.0062379
$a = 9.25$	log 0.9661417
	0.0400961

0.0400961	log 8.6031020 − 10	3.1416	log 0.4971499
180	log 2.2552725	cot 80°	log 9.2463188 − 10
		log e	log 9.6377843 − 10
	0.8583745		
	9.3812530 − 10		9.3812530 − 10

$\phi^\circ = 30.00001$ log 1.4771215 or 30°, which checks

$$\phi^\circ = 30^\circ = 10 \times n = 10 \times 3.$$

Fig. 34 will illustrate graphically this method applied to example.

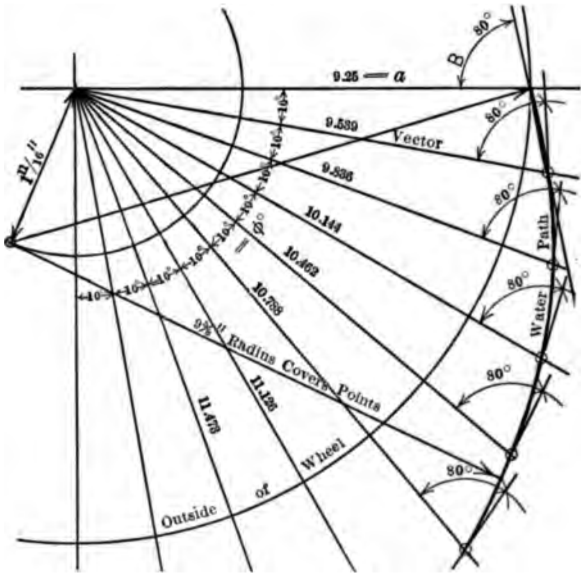


Fig. 34. Graphical Method for Diffusion Vanes.

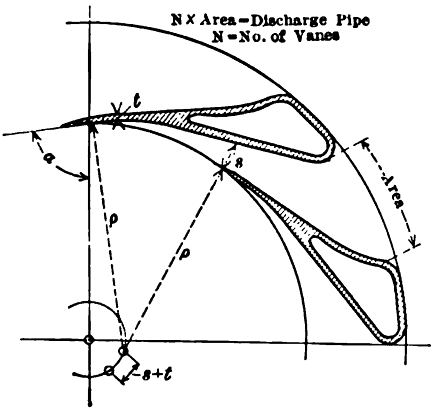
$$\left\{\begin{array}{ll} \log r = \log a + \cot B \times \phi \times \log e, \text{ given } \phi^\circ & \\ \phi = \dfrac{\phi^\circ \times \pi}{180^\circ} & \\ \phi^\circ = \dfrac{(\log r - \log a) \times 180^\circ}{\pi \times \cot B \times \log e} & \begin{array}{c}\phi^\circ \text{ unknown} \\ \text{given } r\end{array} \end{array}\right\}$$

As frequently stated, the object is to utilize the velocity of the water as it issues from the wheel by changing the energy of motion into energy of pressure. To obtain the full benefit of a vortex chamber, the direction of flow of the water must always be the same; in other words, the angle which the water path at any point makes with the corresponding radius must be a constant. Furthermore, the line of direction must be that of the absolute velocity of the water as it leaves the wheel. The equation to this curve, which is a logarithmic, spiral is $r = ae^{m\phi}$ when $\phi = 0$. e = Napierian base, $r = a$. m is the point determined by calculation from the problem. In the ordinary type of turbine pump the water cannot take this path, except

in the last wheel, but must flow around the circumference of the wheel. An easy turn is made from the correct curve. The area of the passageways at the circumference is equal to the area of discharge pipe, or slightly larger, that there may be no sudden velocity changes. In some recent pumps the flow of water throughout the pump from suction to discharge is in a continuous spiral path in order to obviate all shocks and sudden changes. The curves depend, however, upon the fact that the radial flow in the diffusion ring varies inversely as the distance from the center, and the curve of the path is the resultant of the absolute and radial flow. A method of constructing diffusion rings by aid of circles, not governed by any laws, is shown in Fig. 35. The logarithmic spiral curve will give the best efficiency. A simple way to construct curves in the diffusion ring is shown in Fig. 36. Let r_x be any radius of the curve calculated by $r_x = R_e^{\theta \cot \beta}$, where θ in circular measure is $= \frac{\theta° \times 3.1416}{180°}$.

Bisect angle θ by line o^a and prolong. The bisector *cb* is the mean proportional of the radii R and r_x. Erect perpendicular *od*. Describe arc *cdb* on *cb* as a diameter; then *od*, cut off by the arc, is equal to the radius *of*.

By bisecting $\frac{\theta}{2}$ we obtain another point in a similar manner, and thus as many points as are desired may be found.



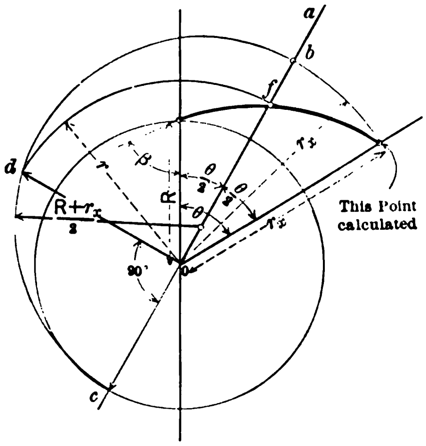


Fig. 35. Fig. 36.

Method of Constructing Diffusion Vanes by Circles.

APPLICATION OF THEORY.

In order to cover the details involved in the mathematical consideration of the problem we will calculate a concrete case, and will consider a pump having 12 stages with approximately 50 feet to the stage. It is now possible

to work single stages as high as 150 feet and even more under proper conditions, but we make the above assumption in order to illustrate

Pump, 4-inch, 12-stage.

Capacity, 150 gallons per minute.
Speed, 1460 revolutions.
Total or useful head, 610 feet.
Diameter of inlet and outlet, 4 inches.
Outside diameter of diffusion ring, 13 inches.
Inside diameter of impeller, $3\frac{7}{8}$ inches.
Diameter of hub of impeller, 2 inches.
Outside opening of diffusion ring, $\frac{1}{2}$-inch wide (assumed).
Outside diameter of impeller, $8\frac{31}{32}$ inches, to suit speed.
Width of opening of impeller at outside, $\frac{3}{8}$ inch.
Inside diameter of diffusion ring, 9 inches.
Angle which tangent to blade makes with radius of outside, or $\theta_2 = 72°$.
Number of blades, 12.

Calculations. — Tangential velocity of outer circumference of impeller

$$= v_2 = \frac{1460 \times 8\frac{31}{32} \text{ inches} \times \pi}{60 \times 12} = \frac{146 \times 287 \times \pi}{2304}.$$

146 log	2.1643529
287 log	2.4578819
π log	0.4971499
	5.1193847
2304 log	3.3624825
$v_2 = 57.13500$	1.7569022 feet per second.

Radial velocity of water at outside,

$$u_2 = \frac{1}{12}\left\{\frac{\dfrac{150 \times 231}{60}}{8\frac{31}{32} \times \frac{3}{8} \times \pi - 12 \times t \times \frac{3}{8}}\right\}.$$

The thickness t of the blades at the outer circumference measures $\frac{3}{8}$, and $12 \times \frac{3}{8} = 4.5$.

The circumference of $8\frac{31}{32} = 28.676$;
of $28.676 - 4.5 = 23.676$, leaving a net length, then

$$u_2 = \frac{\dfrac{150 \times 231}{60}}{\frac{3}{8} \times 23.676 \times 12} = \frac{385}{71.028} = 5.42 \text{ feet per second.}$$

$$\frac{v_2}{u_2} = \frac{57.135}{5.42} = 10.56 \text{ and } \frac{v_2}{u_2} = 10.56 = \text{tang}\,\phi_2 + \tan\theta_2$$

$$\text{tang}\,\theta_2 = \text{tang}\,72° = 3.0777$$

$$\text{tang}\,\phi = 10.56 - 3.0777 = 7.4823,$$

we have $$v_2^2\eta = gH\left(1+\frac{\text{tang }\theta_2}{\text{tang }\phi_2}\right) = gH\left(\frac{\frac{v_2}{u}}{\text{tang }\phi_2}\right)$$

$$v\eta = \frac{gH}{u \text{ tang }\phi_2} = \eta = \frac{gH}{u_2v_2 \text{ tang }\phi_2} = \frac{32.16 \times 610}{12 \times u_2 \times v_2 \times \text{tang }\phi_2}.$$

32.16	log	1.5073160
610	log	2.7853298
		4.2926458
12	log	1.0791812
u_2 = 5.42	log	0.7339993
v_2 = 57.135	log	1.7569022
tang ϕ_2 = 7.4823	log	0.8740351
		4.4441178
η = 0.70555		$\bar{1}$.8485280 = 70 per cent efficiency.

Path of the water or curve in diffusion ring,

r_x = radial distance to any point

$= r_x = r_e \cot\phi_2 \times \phi,$

or $\log r_x = \log r + \phi \times \cot\phi_2 \times \log_e,$

where r = inside radius of impeller = $4\frac{1}{2}$ inches.

ϕ = circular measure of angle between r and r_x (see Fig. 37), in degrees to $\frac{\phi^\circ \times \pi}{180^\circ}$,

e = Napierian base.

r_x is very simply calculated:

(1) $r_{x1} = \log r + \cot\phi_2 \times \phi \times \log_e$.

(2) $r_x = \log r + \cot\phi_2 \times \phi' \times \log_e$.

(3) $r_x'' = \log r + \cot\phi_2 \times \phi'' \times \log_e$.

The difference between (2) and (1) and (3) and (2) is

$$(\phi' - \phi)(\cot\phi_2 \log_e),$$

and by making r_x progress by uniform steps this is a constant.

Make the difference in angles 5°; then

$$\phi = \frac{5 \times x \times \pi}{180} = \frac{x\pi}{36},$$

where x = the multiple of 5 in the angle ϕ. Then

$$r_x = \log r + \cot\phi_2 \times \frac{x\pi}{36} \times \log e$$

$$= \log 4.5 + \cot\phi_2 \times \frac{x\pi}{36} \times \log e$$

$$= \log 4.5 + \left(\frac{\cot\phi_2 \times \pi \times \log e}{36}\right).$$

Note where the diffusion ring is used θ_2 is not so important as without.

$$\text{tang}\,\phi_2 = 7.4823 \quad \log 0.8740351$$

$$\cot \phi_2 = \frac{1}{\text{tang}\,\phi_2} \quad \log \bar{1}.1259649$$

$$\log \cot \phi_2 = \bar{1}.1259649$$

$$\pi \quad 0.4971499$$

$$\log \text{ of } \log e \quad \bar{1}.6377843$$

$$\bar{1}.2608991$$

$$\log 36 \quad 1.5563025$$

$$\frac{\cot \phi_2 \pi \log e}{36} = 0.0050652 \log \bar{3}.7045966$$

$$\log 4.5 \quad 0.6532125$$

	Constant 0.0050652	r_x
$\phi = 0$	$r_x = r \log 0.6532125$	4.5
5°	$r_x = \log 0.6582777$	4.552
10°	0.6633429	4.606
15°	0.6684081	4.660
20°	0.6734733	4.715
25°	0.6785385	4.763
30°	0.6836037	4.826
35°	0.6886689	4.883
40°	0.6937341	4.940
45°	0.6987993	4.999
50°	0.7038645	5.057
55°	0.7089297	5.116

We have simply to add constant above to log of r.

To obtain $r_x =$ for $\phi° = 50°$ we have added the constant ten times:

$$10 \times 0.0050652 = 0.0506520$$

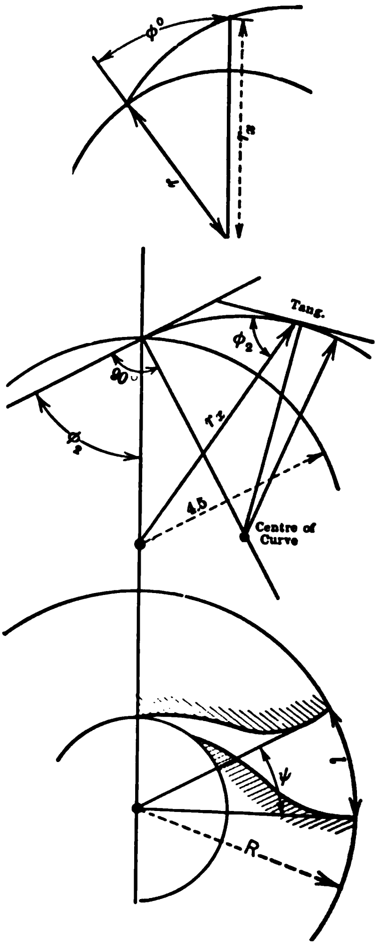
$$\log r = 0.6532125$$

$$\log r_x \text{ for } \phi° = 50 \quad 0.7038645$$

This checks the work.

The next step is to find the arc of a circle which shall pass through the greatest number of points as given by r_x (see Fig. 38). The center of this curve should be on a line perpendicular to the tan at $r = 4.5$, or very nearly so.

Regarding the diffusion ring we may make the following observations: The width, or difference between the inner and outer radii, varies with different builders, as no one can tell the exact



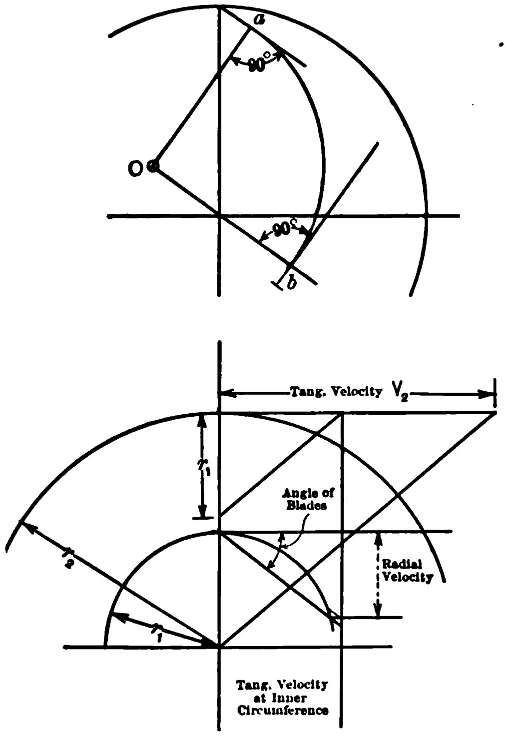
Figs. 37, 38 and 39.
Method of Designing Diffusion Vane.

gain in passing through the ring. It will not do to have too large an angle ϕ_2 or too many guides, as the space becomes so small that the velocity is very great, and it should not be more than the absolute velocity of the water leaving the wheel. Where the water discharges around the ring the curves at the end should be radial as shown in Fig. 39. The angle ψ is easily calculated, thus $l \times w$ when w = width is found; then

$$\frac{\pi \frac{d^2}{4}}{Dw\pi} = \text{ratio of total opening,}$$

with reference to discharge pipe. Here

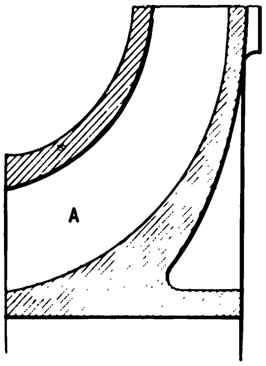
$$D = 13; \quad w = \tfrac{1}{2} \text{ inch}; \quad d = 4 \text{ inches};$$



$$\text{then} \frac{\frac{16}{4}}{13 \times \frac{1}{2}} = \frac{16}{4} \times \frac{2}{13} = \frac{8}{13};$$

$$\text{then} \frac{8}{13} \times 360° = \frac{2880}{13} = 221.50°.$$

We find 6 guides give about the right passage for the velocity and space at the outer circumference; we have, therefore, $\frac{221.5°}{6} = 36.9°$. We will call it,



Figs. 40 and 41.
Method of Designing Impeller Blade.

Fig. 42.
Section of Impeller.

for practical purposes, 35°, and then round the corners. The water will then enter the passage around the wheel at about the same velocity as the water in the passage. When this is not the case there should be a free passage on an Archimedean spiral.

BLADES.

A few remarks only are necessary. To be sure of the inner and outer angles the center O, Fig. 40, should be such as to give a short straight line at both ends, a and C. Strictly speaking, the water should enter the impeller in the direction of the blade at the inner end, and as it is usual to make the radial velocity the same at the inner and outer circumference this angle may be determined as per Fig. 41; but owing to the doubt about the real direction of entrance, this angle is usually made somewhat larger. This is wholly a matter of judgment, as no test can absolutely determine it. Much has been written on it, and, strictly speaking, the blade, having the section A, Fig. 42, should be calculated for each radius, and formulæ have been given for this. But as the calculation is complicated and founded wholly upon assumptions, it is questionable whether it pays to consider this. In fact, most authors assume the tangential velocity of the water at the entrance as zero.

CAPACITY OF CENTRIFUGAL PUMPS.

The velocity of flow through the impeller governs the capacity. When this velocity and the circumferential area, deducting the thickness of vanes, are known, the total capacity can easily be calculated. The speed for a given discharge can be calculated and depends upon the head against which the pump is to work. The water is supposed to rotate in the pump as a solid mass, and delivery commences when the centrifugal force is greater than the total lift including all losses and friction. Let

u_1 = absolute velocity inlet,
u_2 = absolute velocity outlet.

Then the centrifugal force $= \dfrac{u_2^2 - u_1^2}{2g}$. This speed is the maximum, and in practice it will be less, depending on the angle between the impeller vane and the circumference. The efficiency of the pump is also dependent upon this angle. It should be noted that the tip angle has considerable influence upon both the efficiency and the uniformity of power required.

In the formula known as Appold's, $v = 550 + 500\sqrt{Hf}$, Hf is the static head in feet, $= nCf$, in which Cf is the circumference of impeller in feet. 500 is an arbitrary figure but supposed to be equal to $\sqrt{2gh}$ times a constant, and the 550 another arbitrary figure, to give the necessary velocity in feet per minute. This will have to be revised in order to meet later developments in this class of pumps. The entire head produced by a pump depends upon the velocity, disregarding the angles of the vanes. If this velocity $= V$, and total head $= H$, H_v = velocity head, H_f = friction

head, H_s = static head, and V_t = theoretical velocity of H, the equation then becomes

$$V = \frac{V}{V_t}\sqrt{2\,gH} \quad \text{and} \quad V_t = \sqrt{2\,gH}.$$

On closing discharge valve H will equal H_s;

on opening the discharge, $V = \frac{V}{V_t}\sqrt{2\,g\{H_s + H_v + H_f\}}$.

It must not be forgotten that the radial velocity of the water depends upon the dimension of the wheel, and that if the quantity of water pumped is excessive and the radial velocity be increased the frictional losses will also be increased and the efficiency diminished. Some manufacturers, after establishing their sizes and details of design, use arbitrary figures for the velocity in order to obtain the static head; i.e., for H = static head they use a velocity of the outer circumference, $V = 10\sqrt{H}$. Where H is to include friction losses in suction and delivery they use $V = 9\sqrt{H}$. When discharge valve is closed the pressure produced is $H = \frac{V^2}{2\,g}$. In a centrifugal pump the power is directly dependent upon the capacity at a stated speed. The total head is also dependent upon the capacity, therefore the head created can be used as a useful head or lost by closing the valve on the discharge. A partially closed valve causes a lower efficiency, which bears a proportion to the rated efficiency equal to the ratio of the head generated to the total head generated. Assuming a pump working against a head of 250 feet and giving 1000 gallons capacity at 75 per cent efficiency, what would the efficiency be for the same capacity at 175 feet head? The probable efficiency will be $75 \times \frac{175}{250} = 52.5$ per cent, or in other words the horse power can be calculated for the capacity against 175 feet and divided by the brake horse power at 250 feet.

The following empirical formulæ for ready and approximate figuring may be used:

$$\frac{1830 \times \sqrt{H}}{\text{revolutions per minute}} = \text{diameter of impeller in inches.}$$

This will apply to volute pumps.

For turbine pumps with diffusion ring, the formulæ with another constant can be used:

$$\frac{1850\sqrt{H}}{\text{revolutions per minute}} = \text{diameter of impeller in inches,}$$

for all impellers below 6 inches. For 6 inches and larger add $\frac{1}{8}$ inch to the diameter obtained, and for every 3 inches of increase in diameter add another $\frac{1}{8}$ inch.

For dredging work the formulæ become

$$\frac{2000\sqrt{H}}{\text{revolutions per minute}} = \text{diameter of impeller in inches.}$$

CHAPTER X.

SECOND ANALYSIS OR THEORY.

THEORY OF IMPELLERS.

THIS section will treat the subject in a slightly different way to give the designer a method which may be easier and can be more readily applied.

H = head in feet;
Q = capacity in gallons per second or cubic feet per second;
n = revolutions per minute;
u_2 = circumferential speed in feet per second;
v = radial speed in feet per second;
P = power in horse power or foot pounds per second.

α, β, γ are the constants in the general equation, used to show the relation between head, speed, and capacity, or between H, u, and Q.

r, s, t are the constants used in the general equation, which will be used to show the relation between horse power, revolutions, and capacity, or between P, n, and Q.

g = acceleration.

In considering the following, it is well to understand that all curves in the figures submitted are from actual performances, and details have been obtained from working impellers in order to present reliable data on which to base conclusions. Note that in the curves submitted the abscissas show the capacity in gallons, and that the ordinates give the head in feet, from which we get, with a constant speed, what is termed the capacity-head curve. This relation between capacity, head, and speed is expressed in the following equation, which is that of a hyperbolic paraboloid corresponding to the path of the water through the pump. The head due to the centrifugal force will vary with the square of the distance from the center, and the curve assumed by the surface of the water will be that of a parabola,

$$\alpha Q^2 - \beta Qn - \gamma n^2 = -2\,gh,$$

Q, n, and h being variables, and α, β, γ, g, constants.

The relations existing show that the capacity varies directly with the speed, the head being constant, that the head will vary directly as the square of the speed at constant capacity, and that the head will vary directly as the square of the capacity for constant speed. These relations govern all cases and must be clearly understood in designing and operating centrifugal

pumps, and in selecting pumps suitable for a particular purpose. Various curves are illustrated for analyzing the action of the pump.

Fig. 43 shows the relations for constant head.

Fig. 44 shows the relations for constant speed.

Fig. 45 shows the relations for constant speed.

Fig. 46 shows curves for variable speed, with the equations for capacity-head curve and capacity-power curve.

Fig. 47 shows similar curves.

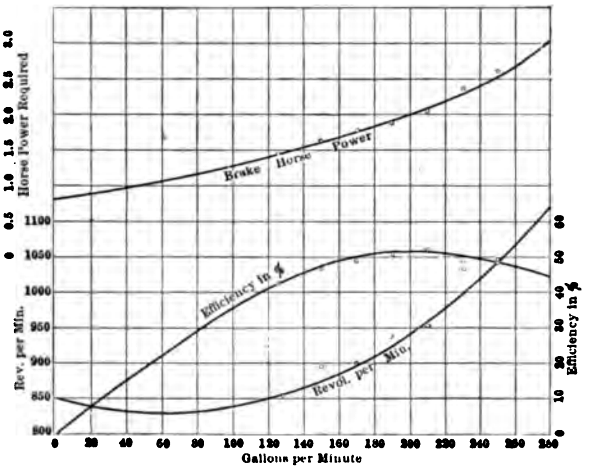


Fig. 43. Characteristics Showing Capacity Relation for Constant Head.

The most common condition is that of constant speed, illustrated in Figs. 44 and 45. These show that at 1150 revolutions the head can vary between 0 and 380 feet, including all friction losses and suction lift. The curve indicated that the largest capacity under no head is 2225 gallons, and under a head of 380 feet 800 gallons, the maximum efficiency of 70 per cent being reached with a capacity of 1200 gallons, 348 feet head. The brake-horse-power curve shows the amount of power required to run the pump at constant speed with a varying capacity and head, the head varying directly as the square of the capacity. The head curve shows that with the discharge valve closed there is a head of 285 feet with no discharge. The capacity increases as the valve is opened, and the head rises until it reaches 380 feet for 800 gallons. At another point where the head is 285 feet a capacity of 1500 gallons is obtained, giving the two limits of operation of the pump.

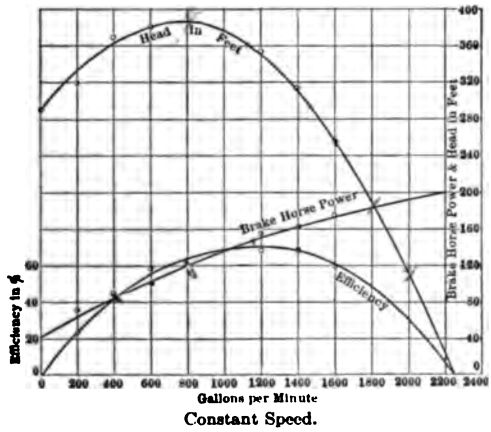


Fig. 44. Characteristics Showing Relations for Constant Speed.

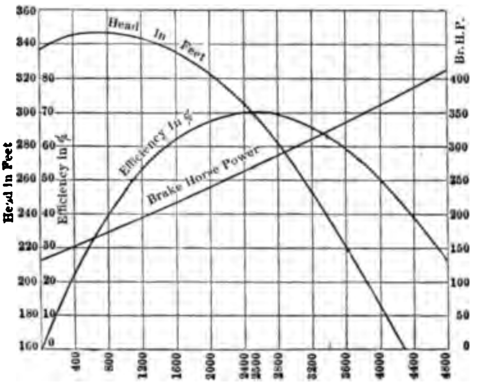
At constant speed and 360 feet head, it would give from 400 gallons to 1150 gallons.

The following conditions should be noted:

First, when capacity Q equal to 0;

Second, when head H equal to maximum;

Third, when head H equal to 0.



Constant Speed.

Fig. 45.

Equation of Impeller:

$Av^2 - Bu_2v - cu_2^2 = -2gh.$

u_2 = circumferential speed, feet per second.

v = radial speed of water on outlet, feet per second.

h = head, in feet.

Example:

Diameter impeller $21\frac{15}{16}$ inches.

$h = 300$ feet.

$n = 900$ rev. per unit ($u = 86$ feet per second).

$$\left.\begin{aligned} A &= 12.17 \\ B &= 1.11 \\ C &= 0.98 \end{aligned}\right\} \text{for 1 impeller.}$$

$$12.17\,v^2 - 1.11\,u_2v - 0.98\,u_2^2 = -2\,gh.$$

Curves A for 1200 r.p.m.
Curves B for 1000 r.p.m.
Curves C for 800 r.p.m.
Curves D for 600 r.p.m.

Equation of Cap.-Head Curve:

$$\alpha \cdot Q^2 - \beta \cdot n \cdot Q - \gamma \cdot n^2 = -2\,gh.$$

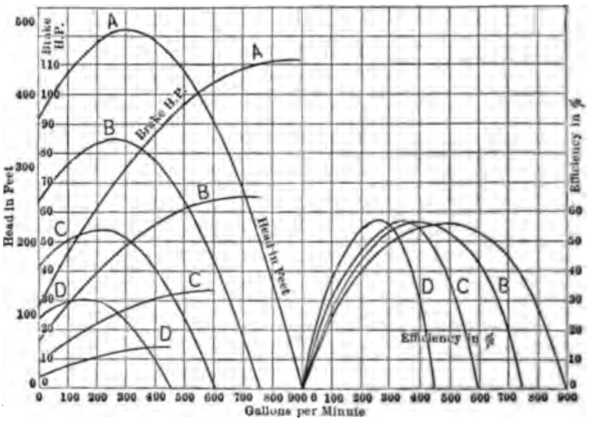
$\alpha = 17455.4.$
$\beta = 19.4335.$
$\gamma = 0.016395625.$

Equation of Cap.-Power Curve:

$$L = r \cdot n \cdot Q^2 + s \cdot n^2 \cdot Q + t \cdot n^3.$$

$r = -108.6466.$
$s = 0.3407783.$
$t = 0.00008699368.$

Q = cu. ft. per sec.
L = sec. foot lbs.
n = r.p.m.

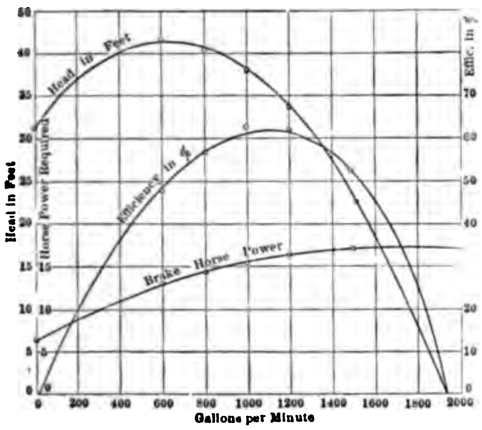


Variable Speed.

Fig. 46.

The maximum head does not occur when the discharge valve is closed, but at a definite capacity differing somewhat in the various types of pumps, but greater in high-head than in low-head pumps, as the volumetric losses in high-head pumps are greater than the hydraulic losses. This is illustrated by considering the pump in question as delivering into a vertical pipe higher than the lift of 380 feet. There would be no discharge, and it

would be impossible to start the column again until the head was reduced to less than 285 feet, or the head which the pump would produce with the discharge closed. The maximum capacity is at zero head. In order to



Constant Speed.

Fig. 47.

Equation of Capacity-Head Curve:

$$\alpha \cdot Q^2 - \beta \cdot n \cdot Q - \gamma \cdot n^2 = -2gh.$$

$\alpha = 323.9365.$

$\beta = 1.6176629.$

$\gamma = 0.005921232.$

For 580 r.p.m.:

$h = -5.036\,Q^2 + 14.587\,Q + 31.$

n = head in feet.

Q = cubic feet.

Equation of Capacity-Power Curve:

$$L = r \cdot n \cdot Q^2 + s \cdot n^2 Q + t \cdot n^3.$$

$r = -0.84913453.$

$s = 0.010273562.$

$t = 0.00001747915.$

For 580 r.p.m.:

H.P. $= -0.89545\,Q^2 + 6.2837\,Q + 6.2.$

L = sec. ft. pound.

H.P. = horse power.

increase this capacity the speed must be increased, as the capacity varies directly with the speed at constant head. With a constant head, and a variable speed and capacity, the efficiency and horse-power curves become

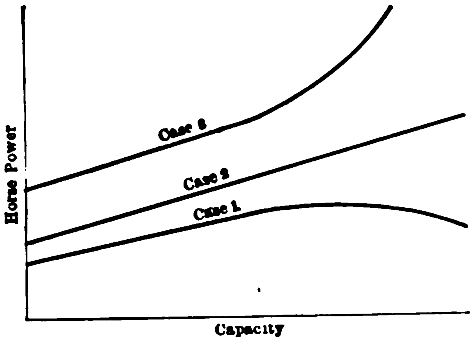


Fig. 48.

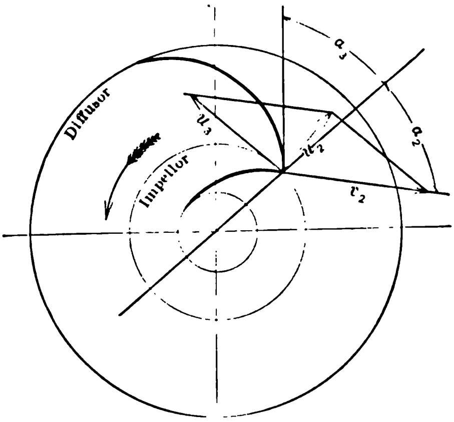


Fig. 49.

Power Curves Showing the Effect of Different Vane Angles.

different, and a point is reached when no increase in capacity can be obtained. The power curve is

$$P = rnQ^2 + sn^2Q + tn^3.$$

The horse power is directly proportional to the square of capacity if the speed is constant, and directly proportional to the cube of revolutions if the capacity is constant. When the discharge valve is closed and there is no delivery, the horse power is directly proportional to the cube of the revolutions. With the speed constant, the equation becomes a parabola, with the capacities as abscissæ and the power as ordinates.

Figs. 48 and 49 show that the variation in the power depends upon the angles. There are three different parabolas, 1, 2, or 3, depending upon the angles. We have case 1 if we select $(\alpha_2 + \alpha_3) < 90$ degrees; case 2, nearly a straight line, if the angles $(\alpha_2 + \alpha_3) = 90$ degrees; and case 3 if the angles $(\alpha_2 + \alpha_3) > 90$ degrees. The equation will solve the question of maximum power at constant speed for the maximum capacity, as $Q_{max}H = 0$.

APPLICATION OF ANALYSIS TO PROBLEM.

The following will illustrate the analysis of any problem.

η = total efficiency of pump;
η_m = mechanical efficiency;
η_v = volumetric efficiency;
η_h = total hydraulic efficiency;
η_{h_1} = absolute hydraulic efficiency;
η_e = efficiency motor;
η_o = over-all efficiency or wire to water.

Total efficiency is the relation between the brake horse power of motor and the actual water horse power.

Total efficiency of pump, $\dfrac{\text{water horse power}}{\text{brake horse power}} = \dfrac{Q \times H \times 8.33 \times 60}{33{,}000}$.

Electric horse power for direct current $= \dfrac{\overset{V}{\text{voltage}} \times \overset{A}{\text{amperes}}}{746}$.

Brake horse power, $\eta_e \times \dfrac{V \times A}{746}$.

η_0, over-all efficiency, $\dfrac{\text{water horse power}}{\text{electric horse power}}$.

Break-horse-power output for alternating-current motors

$$= \frac{\text{volts} \times \text{amperes} \times \sqrt{n} \times \cos\phi \times \eta}{746},$$

where n = number of phases;
$\cos\phi$ = power factor of motor;
η = motor efficiency.

The speed of motors at full load will vary from 2 to 5 per cent on a motor of 200 to 10 horse power, and from 1 to 2 per cent from 500 to 200 horse

power. Small motors of 1 to 10 horse power may vary from 10 to 15 per cent. The ratio between the number of poles and the speeds and cycles is as follows: When f = frequency, p = number of poles. r = number of revolutions; $r = 120\frac{f}{p}$. The slip is expressed in a percentage of the actual revolutions. Actual speed = revolutions (1 − per cent of slip). This slip is due to the resistance opposed to the rotor current.

The elements of total pump efficiency are, therefore, made up of

Mechanical efficiency $= \eta_m$.
Volumetric efficiency $= \eta_v$.
Absolute hydraulic efficiency $= \eta_{ht}$.

Total hydraulic efficiency $= \eta_h = \eta_{ht} \times \eta_v$.
Total pump efficiency $= \eta_m \times \eta_v \times \eta_{ht}$.

The elements of mechanical efficiency are made up of friction of shaft and impeller, etc., in their bearings and is a function of the workmanship and fits. The clearances vary from 0.002 to 0.005 inch, and should never be larger than 0.006 inch. The volumetric efficiency, η_v, expresses the ratio of the amount of water entering the pump to that which is discharged, the loss being due to leakage in the running fits. The greatest losses in a turbine pump occur between the impeller and the diffusion ring. These vary between 2 to 10 per cent. *The absolute hydraulic efficiency* η_{ht} expresses the ratio between the useful head and the total pumping head. If h represents the former and H the latter, then the total head H is made up of h and all frictional and shock losses of water in the pump, and is the most important factor to consider in designing pumps. Information relating to these losses is meager and incomplete. Another serious loss is due to skin friction between the impeller and the water. The skin friction increases with head pumped against more rapidly than the head increases with respect to the velocity. The head against which the pump operates varies as the square of the velocity and the wasted power as the cube of the head, or H^3. The work lost in disk or skin friction varies as the square of the radius, therefore a small impeller at a high number of revolutions will waste less power than a large one running slower, both having the same peripheral velocity. *The largest losses by surface friction* occur along the walls of the casings, and if the surfaces of the stationary walls and impellers were alike the water would have a rotating motion at a speed one-half that of the impeller. This loss may be reduced by having the walls smooth and the impellers polished and by having proper clearance between impellers and casings. *Experiments have been made abroad* on the power loss by skin friction, and the following formula has been obtained:

$$W = F \times \frac{1}{n^2} \times h^{2.5} \text{ foot lbs.}$$

W = power due to resistance of rotating disk in foot lbs.;
F = constant 8132;
n = revolutions per minute;
h = head in feet.

In addition to the surface friction of the water, there are losses due to molecular friction. The head is shown to be directly proportional to the square of the revolutions and the power lost on account of friction to some power of revolutions. The results show that these losses are greater in high-head pumps than in low-head pumps.

In designing we may expect a total efficiency of 90 per cent in the large pumps, under favorable conditions, with a lower efficiency in the smaller ones. It is absolutely necessary to select the right efficiency when designing this class of pumps, and this can be obtained from tests.

The curve of relationship between the capacity and efficiency is a parabola commencing at zero, its vertex showing the maximum efficiency, and coming down again to zero as the head approaches zero. The maximum point or vertex should occur under the conditions for which the pump is designed.

The general equation for the impeller is

$$Av^2 - Bu_2v - Cu_2^2 = -2\,gh,$$

where

u_2 = circumferential speed of impeller;
v = radial speed of water on outer circumference of impeller;
g = acceleration;
h = head.
A, B, C are constants.

This equation is analogous to the capacity-head curve, v being directly proportional to the capacity

$$F \times v = Q.$$

Q = capacity; F = sectional area.
u_2 is directly proportional to the speed

$$u = \frac{2\,r\pi u}{60}.$$

The general equation for capacity head is a hyperbolic parabola,

$$\alpha Q^2 - \beta uQ - \gamma u^2 = -2\,gh,$$

where Q, h, and u are variables.

If n is constant it becomes a parabola as illustrated in Fig. 46 on page 50. The figure shows the hyperbolic paraboloids for speeds at 600, 800, 1000, and 1200 revolutions. Fig. 50 illustrates the curve with a constant head and the capacity and speed as variables. It represents an actual performance under test. The equation $\alpha Q^2 - \beta uQ - \gamma u^2 = -2\,gh$ becomes a hyperbola. The figure shows the relations between these hyperbolæ and the

capacity and speed at constant heads of 20, 25, 35, and 45 feet. Fig. 51 shows the same condition of a larger pump operated under constant head and a variable speed and capacity.

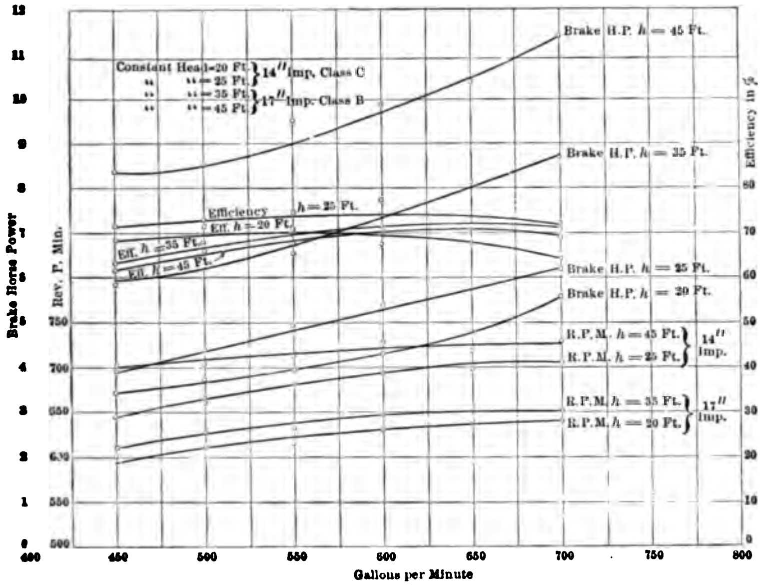


Fig. 50. Curves of Relation between Capacity and Efficiency.

As an example, Fig. 52 will illustrate the method used in designing impellers.

General equation,

$$Av^2 - Buv - Cu^2 = -2gH,$$

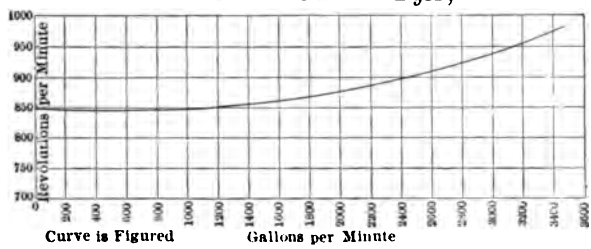


Fig. 51. Curves of Relation between Capacity and Efficiency.

A, B, C being constants or coefficients. H = head. $2g = 64.4$. For v and u see Fig. 52.

$$v = \text{feet per second} = \frac{\text{capacity in cubic feet per second}}{(D_2 \times \pi - s' \times z)b \text{ in square feet}};$$

$$u = \text{feet per second} = \frac{D_2^{\text{in }ft} \times \pi \times \text{revolutions per minute}}{60}.$$

The equation

$$Av^2 - Buv - Cu^2 = -2gH$$

represents a hyperbolic paraboloid, and by taking u, v, or H as constants the three characteristic curves of the pump can be obtained, namely: Revolutions constant, capacity and head varying; H constant, capacity and revolutions varying; capacity constant, head and revolutions varying.

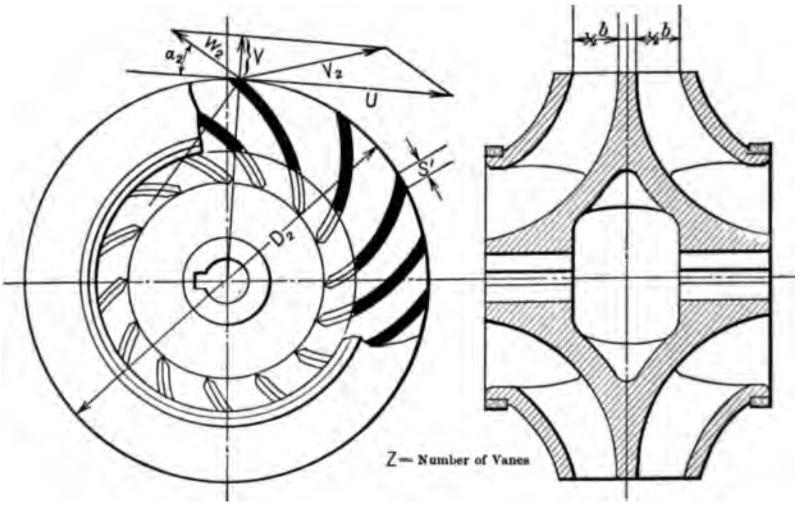


Fig. 52. Double Suction Impeller.

A, B, C depend upon the design of pump and impeller, and can be found from previous available tests, or approximately from the design of the impeller. The impeller shown in Fig. 52, for a 66-inch pump, gives the following values for $A = 4.15$; $B = -0.17$; $C = 0.98$, making the equation read

$$4.15\,v^2 + 0.17\,uv - 0.98\,u^2 = -64.4\,H.$$

Fig. 53. Equation Curves for Impellers.

In order to determine these constants from the fundamental equation, which contains the variables Q, the capacity; u, the revolutions; and H, the head, one can be taken as constant and we can obtain three curves as per Fig. 53. These are the characteristic curves for constant head, con-

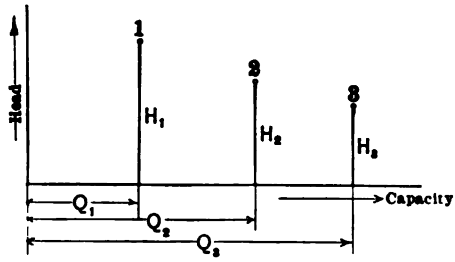


Fig. 54. Readings to Determine Equation Curves.

stant speed, and constant capacity. The values of A, B, and C are usually, however, determined by tests, for which the following readings are required as per Fig. 54. Points 1, 2, 3 are test readings with respective heads H_1, H_2, H_3, and capacities Q_1, Q_2, and Q_3. Introducing these values into the equation given, we have

$$AQ_1 - BQu - Cu^2 = -2\,gH_1,$$
$$AQ_2 - BQ_2u - Cu^2 = -2\,gH_2,$$
$$AQ_3 - BQ_3u - Cu^2 = -2\,gH_3.$$

The unknown constants A, B, C can then be obtained.

The values can be determined without the aid of tests, but it is necessary to substitute in place of capacity Q the relative velocity w_2

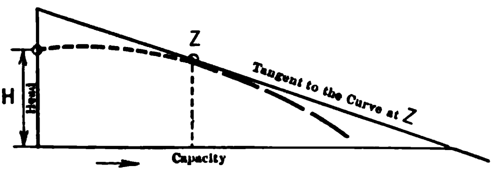


Fig. 57. Method for Finding Characteristics.

of water leaving impeller, and in place of u, the revolutions, the circumferential speed V_5 or u_2, making the equation as follows:

$$A - w_2^2 - Bw_2u_2 - Cu_2^2 = -2\,gH.$$

w_2 and u_2 are in direct proportion to Q and u, therefore, referring to Figs. 68 and 69,

$$A = \Big\{1+\zeta - \phi\cos^2\alpha_5\Big\{\left(\frac{Fw_2}{Fv_2}\right)^2 + (\phi-1)\Big\}\left\{\left(\frac{Fw_2}{Fw_1}\right)^2 + \left(\frac{Fw_2}{Fs}\right)^2\right\}$$
$$+(1+\zeta)+(\lambda+\zeta)\left(\frac{Fw_2}{Fv}\right)^2 - \phi\cos(\alpha_1+\beta)\frac{Fw_2}{Fv_2},$$

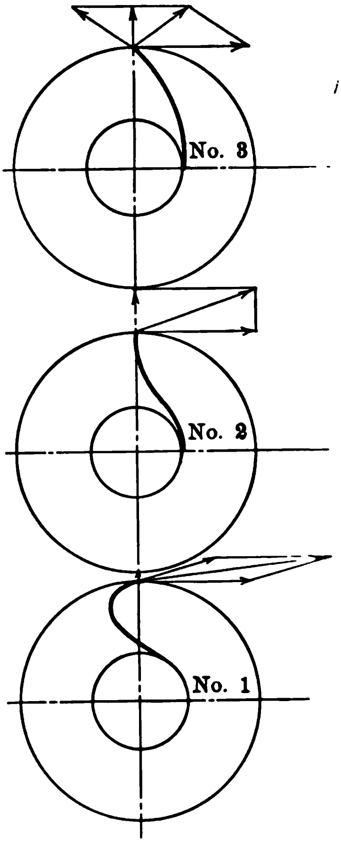


Fig. 55. Various Forms of Impellers for Different Commercial Problems.

$$B = \phi\left\{2\frac{r}{R}\frac{Fw_2}{Fv_2}\sin\alpha_5\cos\alpha_5 + \sin\beta\frac{Fw_2}{Fs}\right\},$$
$$C = 1 + \{\phi\sin^2\alpha_5 - 1\}\left\{\frac{r}{R}\right\}^2.$$

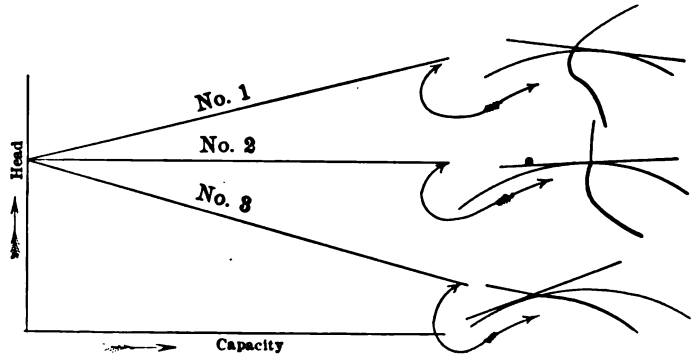


Fig. 56. Various Forms of Impellers for Different Commercial Problems.

ζ = coefficient of friction along vanes, approximately 0.1;
ϕ = coefficient of losses due to shocks, approximately 1.2;

$$\lambda = \frac{2g(1-\eta)H}{v^2};\ v = \text{discharge}$$

velocity; Fv = discharge area; η = hydraulic efficiency.

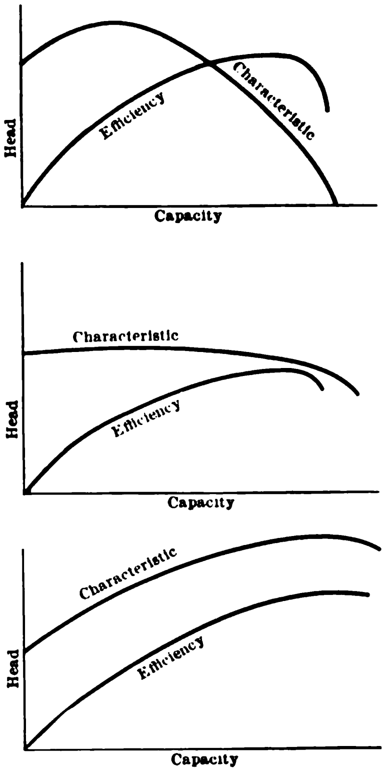


Fig. 58. Power Characteristics for Different Shaped Vanes of Impellers.

SHORTER METHOD OF FINDING CHARACTERISTICS.

A shorter method of finding the characteristics is as follows: Take H' as the head with the discharge valve closed, which by previous equation is $H' = \mu \frac{u_2^2}{2g}$; μ = varying between 0.9 to 1.1, according to the proportions of impeller. The ordinary equation for a centrifugal impeller is

$$u_2 = \sqrt{\frac{1}{\eta}}\sqrt{(1+\text{cotg}\,\beta\,\text{tang}\,\alpha_1)gH.}$$

The direction of the lines changes according to angle α_1 (see Figs. 55 and 56). Assuming that we intend to figure an impeller for condition Z given in Fig. 57, and knowing the head H_1, we can plot the curve from the three elements, two points and a tangent. Referring to Figs. 55, 56, and 58, the various forms of impeller vanes show the different characteristics found in the usual type of commercial pump. In No. 1 the power increases considerably, over the

limit, and this form is limited to conditions where the load is variable, due to frictional resistances other than the regular pumping head. No. 2 will produce a constant head for best efficiency, but has the disadvantage that the power increases rapidly. No. 3, with a constant speed, will maintain a uniform efficiency under a varying head.

The extent to which a designer can go in the development of the shape of the vanes depends upon the conditions of service, and should be closely investigated.

Fig. 59 shows the curves described.

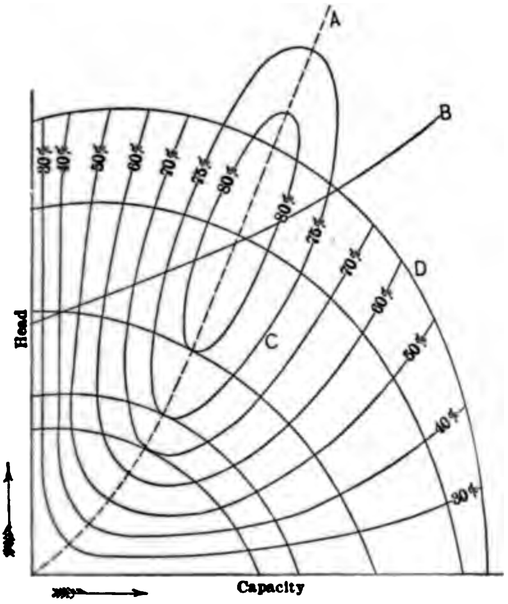


Fig. 59. Characteristics for Curves.

Curves *A*. Connecting Points of Highest Efficiencies Speed or Revolution Constant.

Curves *B*. Connecting Points by Highest Efficiencies Capacity Constant.

Curves *C*. Connecting Points of Equal Efficiencies.

Curves *D*. Showing Head for Constant Revolutions.

CHAPTER XI.

GRAPHICAL ILLUSTRATION FOR DETERMINING THE IMPORTANT ANGLES.

It is advisable to recall the relations existing in plane trigonometry, and the measuring of angles in degrees and minutes in circular measure. Lines having the names sines, cosines, tangents, and cotangents bear a fixed relation to each other for any given angle and radius. Fig. 60 gives these ratios.

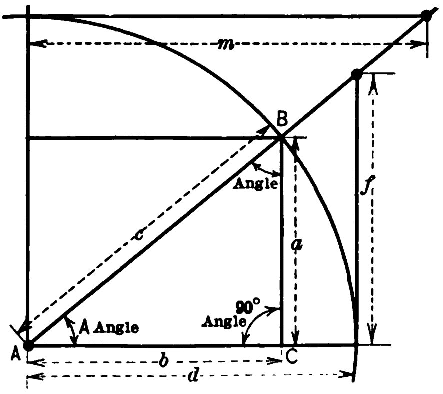


Fig. 60.

Relation of Angles.

$$\text{sine } A = \frac{a}{c} \quad \text{sine } B = \frac{b}{c} \quad \text{hence } a = c \sin A, \quad b = c \sin B$$

$$\cos A = \frac{b}{c} \quad \cos B = \frac{a}{c} \qquad b = c \cos A, \quad a = c \cos B$$

$$\text{tang } A = \frac{a}{b} \quad \text{tang } B = \frac{b}{a} \qquad a = b \,\text{tang} A, \quad b = a \,\text{cotg} A$$

$$\cot A = \frac{b}{a} \quad \text{cotg } B = \frac{a}{b} \qquad b = a \,\text{tang} B, \quad a = b \,\text{cotg} B$$

Therefore

$\sin A = \cos B$ $\quad a =$ is the sine

$\sin B = \cos A$ $\quad b =$ is the cosine

$\text{tang } A = \cot B$ $\quad f =$ is the tangent

$\text{tang } B = \cot A$ $\quad m =$ is the cotangent

The general equation as obtained in the first part of the analysis gives

$$V_5 = \sqrt{\frac{gH}{\eta_h}\left(1 - \frac{\text{tang}\,\beta_0}{\text{tang}\,\alpha_0}\right)}$$

or

$$\frac{\eta_h V_5^2}{gH} = 1 - \frac{\text{tang}\,\beta_0}{\text{tang}\,\alpha_0} \quad \text{or} \quad \frac{1}{\text{cotg}} = \text{tang, Fig. 61,}$$

or

$$= 1 - \frac{\text{cotg}\,\alpha_0}{\text{cotg}\,\beta_0} \quad \text{or} \quad \frac{\text{cotg}\,\alpha_0}{\text{cotg}\,\beta_0} = 1 - \frac{V_5^2\eta_h}{gH} \text{ No. 1.}$$

This gives an equation for finding angle α_0. It can also be found by taking the angle β between the vertical line and the absolute velocity of water (see Fig. 62) and α_1, the angle between the vertical line and the relative velocity of water. Then

$$\frac{\text{tang}\,\alpha_1}{\text{tang}\,\beta} = \mp\left(1 - \frac{V_5^2\eta_h}{gH}\right) \quad \text{and} \quad \text{tang}\,\beta = \mp\,\text{tang}\,\alpha_1 = \frac{V_5}{V_1}.$$

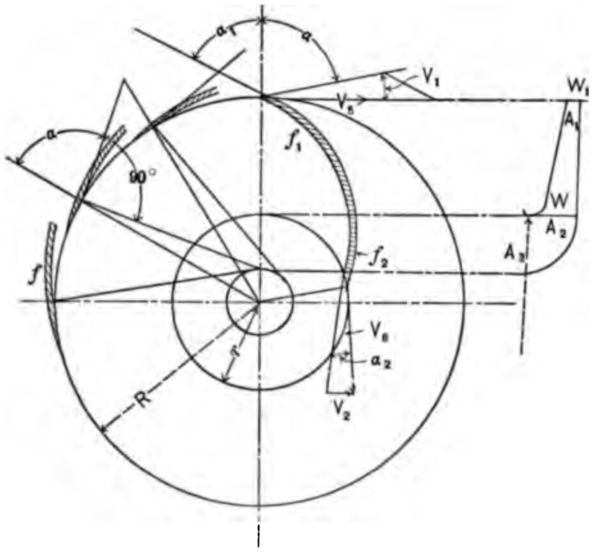


Fig. 61. Diagram for Angles.

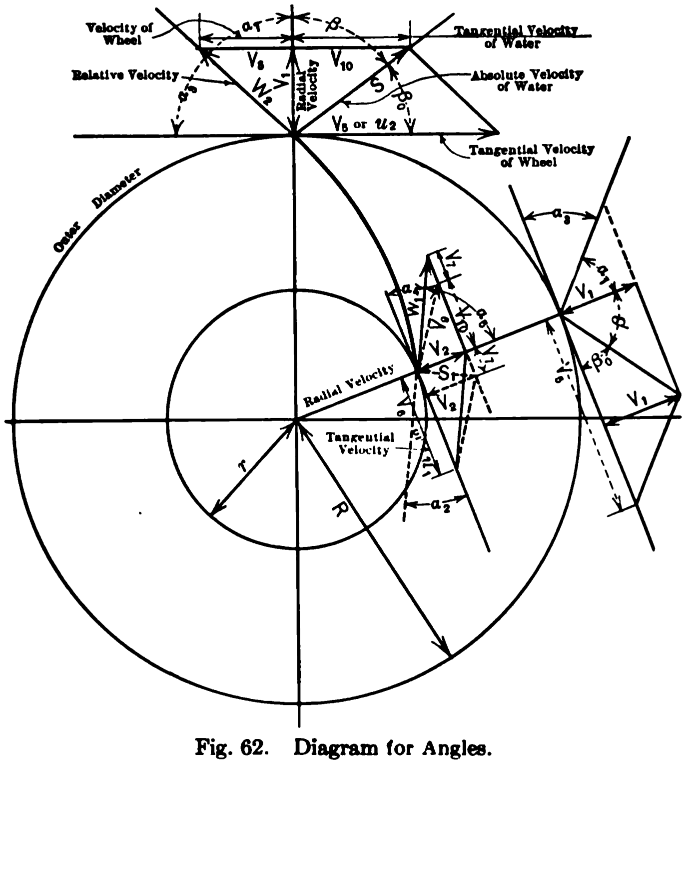


Fig. 62. Diagram for Angles.

α_1 = angle between radial and relative velocity.
$\alpha_2 = 90° - \alpha_1$.
$\alpha_0 = \alpha_1 + \beta + \beta_0$ = angle between tangential and relative velocity.
β = angle between radial and absolute velocity.
β_0 = angle between tangential and absolute velocity.
α_4 =
α_5 =
V = velocity of water in discharge.
V_1 = radial velocity water at outlet in wheel.
V_2 = radial velocity water at inlet in wheel or S_1.
V_5 = tangential velocity also used as u_2 at outlet.
V_6 = tangential velocity also used as u_1 at inlet.
V_7 = velocity of whirl at inlet.
V_8 = velocity of whirl at outlet.
V_9 = relative inlet speed also W_1.
V_{10} = tangential velocity of water.
S = absolute outlet speed.
S_1 = absolute inlet speed considered also as V_2 to prevent complication, assuming water enters radially at entrance without shock.

The negative sign is used for α_1 and β when they are on different sides of center line, and the plus sign when on same side (see Fig. 64).

Referring again to Fig. 62, having found the angle α_0, it remains to find angle β_0.

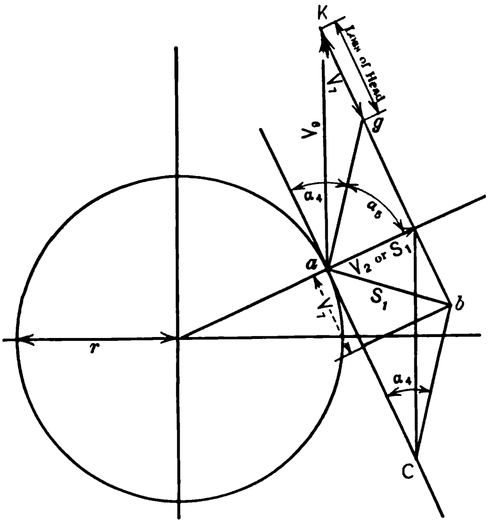


Fig. 63. Diagram for Angles.

de = absolute speed of water leaving impeller;
ef = relative speed of water leaving impeller.

$$\frac{de}{\sin \alpha_0} = \frac{ef}{\sin \beta} = \frac{V_5}{\sin (\alpha_0 - \beta_0)}$$

(see Fig. 65).

V_1 = radial speed = $de \sin \beta_0$,

and by this we obtain

$$V_1 = V_5 \frac{\sin \alpha_0}{\sin (\alpha_0 - \beta_0)} \times \sin \beta_0,$$

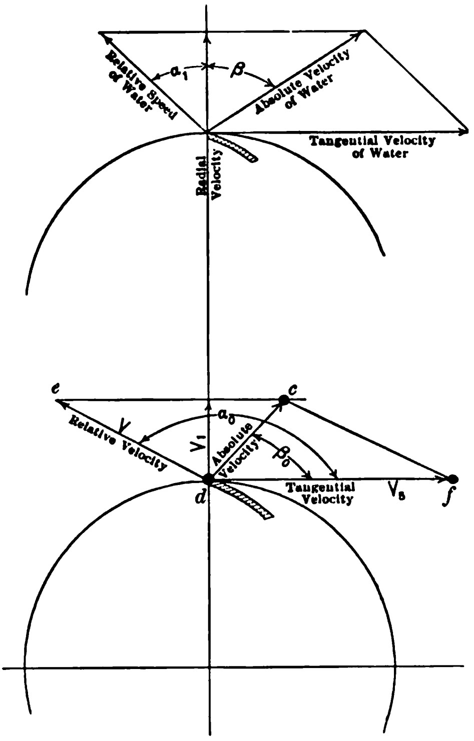
$$V_1 = V_5 \frac{1}{\operatorname{cotg} \beta_0 - \operatorname{cotg} \alpha_0},$$

$$\text{or } \operatorname{cotg} \beta_0 - \operatorname{cotg} \alpha_0 = \frac{V_5}{V_1} \text{ No. 2.}$$

This equation, together with the one for α_0, will solve the angle β_0. For simplicity in solving Fig. 63, it is assumed that the water enters the impellers without loss and radially, hence V_2 is equal to s_1.

The water entering the impeller has a radial velocity V_2 and inner part of the impeller a circumferential velocity of V_6. Let the outer part of the

impeller have a circumferential velocity V_5, and let the velocity of whirl at entrance be V_7 and at outer circumference V_8. Placing V_2 radially and V_7 tangentially a parallelogram is obtained, *ab* denoting the velocity at entrance, and by making *ac* equal to the tangential velocity, we obtain *bc*, the velocity of the water relative to the impeller. This relative velocity determines the entrance angle, as the vane must be tangent to it. For the outer diameter, making the velocity of whirl V_8, the velocity of wheel V_5, and the radial velocity V_1, the outer relative velocity *ef* can be ascertained, which determines the direction of the vane at outer circumference. The work done by the water through the impeller is $\frac{1}{g}(V_8 \times V_5 - V_7 \times V_6)$ foot pounds per pound of water,



Figs. 64 and 65. Velocity Diagram.

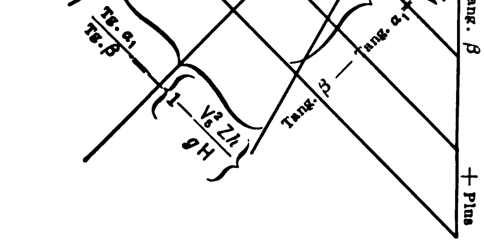


Fig. 66. Graphical Diagram for Angles.

or assuming a radial entrance the expression becomes $\frac{V_8 \times V_5}{g}$.

Assuming water on entering is turned from a radial direction into one of rotation, that its velocity is V_2 and the direction of vanes is V_9, the water would enter without shock. After entering the actual velocity becomes *ab*. The change of velocity relative to impeller is from *ak* to *ag*, and the loss of head is $\frac{(ag \times ak)^2}{2g} = \frac{(V_2 \times ak - V_2 ag)}{2g}$ $= V_6 - V_2 \cot \alpha_4 =$ loss of head at entrance.

Calling this $L = \frac{1}{2g}(V_6 - V_2 \cot \alpha_4)^2$ (see Fig. 63).

GRAPHICAL CHART FOR QUICK REFERENCE.

This can be laid out graphically (see Fig. 66), by using the abscissas for tang α_1 and tang β as ordinates.

Equation No. 1 will give straight lines, beginning at 0, for different values of the tangential velocity V_5, the efficiency η_h, and the total head H.

Equation No. 2 will give parallel lines when less than an angle of 45 degrees with the coördinate axis.

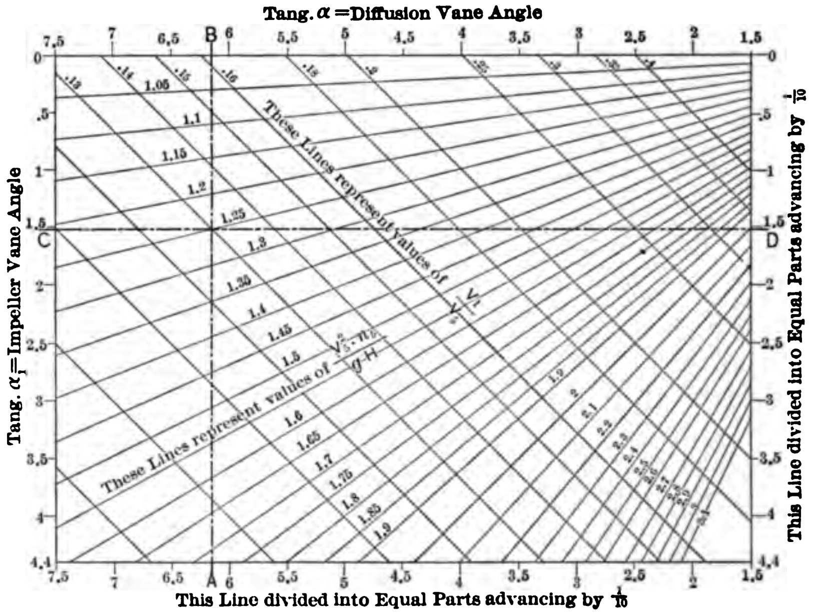


Fig. 67. Graphical Chart for Angles.

The method to be adopted is to choose a tangential velocity according to the speed and to select a convenient diameter for the impeller, thus obtaining a value for $1 - \frac{V_5{}^2 \eta_h}{gh}$, which will give one of the lines radiating from 0. Furthermore, we have the radial velocity $V_1 = \frac{Q}{\pi \cdot 2\,R \cdot W_1}$; W_1 being width of impeller at the outside, Q being capacity in cubic feet or gallons per second, R = radius of the outside of impeller. When we have obtained values for $\frac{V_5}{V_1}$ we will have one of the parallel lines. The intersection between the radiating line and 45-degree line gives the two angles α_1 and β as shown in Fig. 66.

A chart can be made for ready reference in order to determine graphically the diffusion vane and impeller angles as shown in Fig. 67.

Parallel lines with a value of $\frac{V_1}{V_5}$ and intersecting lines with a value $\frac{V_5^2}{gH}$ will give tang α by projecting parallel to A–B, for α by projecting parallel to C–D.

Example. See equation No. 1.

$$\frac{V_5^2\eta_h}{gH} = \frac{3504 \times 0.80}{32.2 \times 64} = 1.36,$$

where head 64 feet $= H$; $V_5 = 58$ feet; $\eta_h = 0.80$; $V_1 = 65$ feet.

$$\frac{V_1}{V_5} = \frac{6.5}{58} = 0.112.$$

If the diagram were laid out the line 1.36 would intersect the $\frac{V_1}{V_5}$ line 0.112, and the tangent of α would be 6.65 and of α_1 2.4. Hence $\alpha = 81.5°$ and α_1 67.5°.

Angle $\alpha_3 = 90° - 67.5° = 22.5°$ (see Fig. 68), and for the diffusion ring the angle would be $90° - 81.5°$, or 8.5°.

METHOD OF CORRECTING IMPELLER-VANE ANGLE.

Assuming a pump designed for a head of 170 feet which on being tested gave an actual head obtained on test of 157 feet, how much must the vane angle be increased in order to get the required head? (See Fig. 68.)

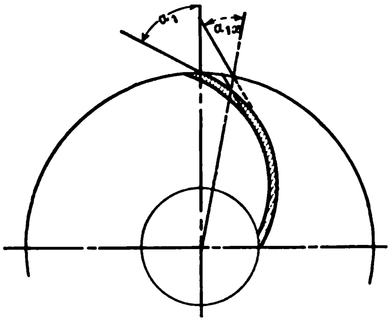


Fig. 68. Correcting Angle at Tip of Vane.

tang α_{1x} being the new angle $= \frac{157}{170}(\text{tang}\,\alpha + \text{tang}\,\alpha_1) - \text{tang}\,\alpha$.

Assuming $\alpha = 74°$ and $\alpha_1 = 63°$,

$$\frac{157}{170}(3.48 + 1.96) - 3.48 = 5.02 - 3.48 = 1.54.$$

tang $\alpha_{1x} = 1.54$ or 57°, or it must be raised the difference between 63° and 57°, or 6°.

CHAPTER XII.

THIRD ANALYSIS OR THEORY.

THEORY OF IMPELLERS.

H = head in feet.
Q = capacity in gallons or cubic feet per second.
n = number of revolutions per minute.
$2R$ = outside diameter in feet.
$2r$ = inside diameter in feet.
V_5 = tangential velocity at outer circumference in feet.
V_6 = tangential velocity at inner circumference in feet.
η_{h_1} = absolute hydraulic efficiency or the ratio between the useful head h and the total pumping head H.
g = acceleration, or 32.2.
α = diffusion-vane angle at outer circumference of impellers.
α_1 = impeller-vane angle at outer circumference of impellers.
α_2 = impeller-vane angle at inner circumference of impellers.
W = width of impeller at hub.
W_1 = width of impeller at circumference.
A = area of discharge pipe in square feet.
A_1 = area of impeller at outer circumference in square feet.
$A_1 = 2R \times \pi \times W_1$.
A_2 = area of impeller at inner circumference in square feet.
$A_2 = 2r \times \pi \times W$.
A_3 = area at hub of impeller, the latter being C in diameter.

$$A_3 = 2r^2\frac{\pi}{4} - C^2\frac{\pi}{4} \text{ in square feet.}$$

A_4 = area of suction pipe in square feet.
Z = number of diffusion vanes.
Z_1 = number of vanes in impeller at outer circumference.
Z_2 = number of vanes in impeller at inner circumference.
It is customary to make $Z_1 = 2Z_2$.
f = thickness of diffusion vanes.
f_1 = thickness of impeller vanes at outer circumference.
f_2 = thickness of impeller vanes at inner circumference.
V = velocity of water in discharge pipe.
V_1 = radial velocity in section A_1 of impeller.
V_2 = radial velocity in section A_2 of impeller.
V_3 = velocity in section A_3.
V_4 = velocity in suction pipe.

To prevent shocks and losses, an impeller must be so designed that the velocities will increase gradually in going from section A_3 to A_1.

The most important values required are the inner and outer vane angles, which may be determined by the following equations:

$$V_5 = \sqrt{\frac{1}{\eta_{h_1}}}\sqrt{(1 + \operatorname{cotg}\alpha \times \operatorname{tang}\alpha_1)\, gH,}$$

or

$$V_5 = \sqrt{\frac{gH}{\eta_{h_1}}(1 + \operatorname{cotg}\alpha \times \operatorname{tang}\alpha_1)},$$

and

$$H = \frac{V_5^2 \eta_{h_1}}{g\,(1 + \operatorname{cotg}\alpha \operatorname{tang}\alpha_1)},$$

$$\operatorname{cotg}\alpha \operatorname{tang}\alpha_1 = \frac{V_5^2 \eta_{h_1}}{2\,gH} - 1.$$

The angle α is usually between 70 and 83 degrees, depending somewhat upon the capacity.

The absolute hydraulic efficiency to be selected is between 69 to 80 per cent, depending upon whether the pump is of the volute or diffusion type, and also upon the head, as it governs the diameter of the impeller.

The angle α is such that the velocity V_5 is larger than V_2, but care must be taken that the space W_1 does not, with small angles, become too small. When the value of α has been calculated we have

$$\operatorname{tang}\alpha_1 = \frac{V_5 \eta_h \operatorname{tang}\alpha}{gH} - \operatorname{tang}\alpha,$$

or

$$\operatorname{tang}\alpha_2 = \frac{V_6}{V_2} = \frac{V_5}{V_2}\frac{r}{R}.$$

Example No. 1. — Capacity, 25,000,000 gallons per 24 hours; head, 200 feet. Dividing this head into two stages of 100 feet each, we find as follows: *Size of suction and discharge:* Selecting a water speed in feet per second from 6.6 to 8.2 feet, we will find

$$A \text{ and } A_4 = \frac{2325}{60 \times 8.2} \text{ to } \frac{2325}{60 \times 6.6} \text{ square feet,}$$
$$= 4.72 \text{ to } 5.87 \text{ square feet,}$$
$$= 29.4 \text{ to } 32.8 \text{ inches diameter.}$$

We therefore select a 30-inch discharge, and as it is desirable to have the suction pipe a size larger a 36-inch suction is chosen in order to cut down the friction and other losses.

To find the inside diameter of impeller, or $2\,r$: The velocity of the water entering the impeller at A_3 is V_3 can be determined by the empirical formula, $V_3 = 0.09$ to $0.12\,\sqrt{2\,gH}$, in which $g = 32.2$ and $H = 100$ feet.

V_3 becomes 7.23 feet to 9.64 feet per second.

Selecting 9.2 feet per second and remembering that V_3 must be larger than V_4, we find that

$$\operatorname{cotg}\alpha \times \operatorname{tang}\alpha_1 = \frac{V_5^2 \times \eta_h}{2\,gH} - 1.$$

Taking tang α = 4.2, we obtain

$$\text{tang}\,\alpha_1 = \frac{79.1^2 \times 0.83 \times 4.2}{32.2 \times 100} - 4.2, \text{ or } 2.8,$$

$$\frac{1}{\text{cotg}} = \text{tang}.$$

This angle gives the radial velocity V_1 of 11.5 feet per second of the water at the circumference of the impeller, which is satisfactory, as it is larger than V_2 and V_3.

Width of impeller at circumference, W_1, and at hub = W.

In order to determine this the impeller should be laid out on the drawing board and from the information already obtained the internal passages can be properly determined, and with the selection of the number of vanes the space can be calculated. In laying the impeller out it is well to allow 10 per cent for losses in the passage through it. The angle α_1 of the impeller at inner circumference can be obtained as follows:

$$\text{tang}\,\alpha_2 = \frac{V_6}{A_3} = \frac{39.6}{10.8}, \text{ or } 3.66,$$

where A_3 = 10.8 feet per second,

and V_6 = 39.6 feet per second.

The area at A_3 will be $\frac{2325}{60 \times 9.2}$ = 4.2 square feet; adding 0.5 square foot for the hub, it becomes 4.7 square feet. The diameter of impeller at the inside is $\sqrt{\frac{4.7}{\frac{\pi}{4}}}$, or 2 feet 6 inches diameter. This would give a very slow speed and would require a large wheel, hence it would be best to increase the velocity at the entrance of impeller to 10.8 feet per second, which would give $A_3 = \frac{2325}{60 \times 10.8}$ = 3.6 square feet, plus 0.5 square foot, or 4.1 square feet in all, or a diameter of 2 feet $3\frac{1}{2}$ inches. It is recommended that the outside diameter of the impeller $2R = 4r$, which in this case would give $2R$ = 4 feet 7 inches diameter, or R = 2 feet $3\frac{1}{2}$ inches.

The circumferential velocity of the impeller:

$$\text{This being } V_5 \cong \sqrt{\frac{1.4\, g \cdot H}{1 - \left(\frac{2\,r}{2\,R}\right)^2}} = \sqrt{\frac{1.4 \times 32.2 \times 100}{1 - (0.5)^2}} = 77.6 \text{ feet per second.}$$

The number of revolutions would then become

$$n = \frac{60 \times V_5}{\pi \cdot 2\,R} = \frac{60 \cdot 77.6}{\pi \cdot 4.58} = 325 \text{ revolutions per minute.}$$

Calling this 330 revolutions, the speed V_5 becomes 79.1 feet per second, and $V_2 = \frac{V_5}{2}$ = 39.6 feet per second.

The angles of vanes may be calculated as described on pages 63 and 64, Chapter XI.

α = diffusion-vane angle;

α_1 = impeller-vane angle;

η_h = absolute hydraulic efficiency, assumed at 83 per cent.

The method thus described, with the velocity diagram, Chapter XI, will give the principal details governing the calculation of a turbine pump. The shape of the vanes must be a continuous curve. In order to start the pump against a full head the velocity of impeller V_5 must be larger than

$$\sqrt{\frac{1.35 \text{ or } 1.4 \times 32.2\, H}{1-\left(\frac{2\,r}{2\,R}\right)^2}}.$$

APPLICATION OF ANALYSIS TO PROBLEM.

Example No. 2. — 8-inch 3-stage turbine pump.

Conditions. — 1800 gallons or 241 cubic feet per minute;
250 feet head, or 83.3 feet per stage;
800 revolutions per minute.

Volumetric efficiency, 95 per cent; total, 65 per cent.

Hydraulic efficiency, $\frac{65}{95}$ = 68 per cent.

Diameters of impellers: hub, $2\frac{1}{2}$ inches, assumed; inside diameter, $2\,r = 6\frac{1}{2}$ inches, assumed; outside diameter, $2\,R = 18\frac{3}{4}$ inches, assumed.

The outside diameter gives a speed of

$$V_5 = \frac{\pi\left\{\frac{18.75}{12}\right\}800}{60} = 65 \text{ feet per second.}$$

Speed required to start pump against full head,

$$V_5 > \sqrt{\frac{1.4 \times g \times H}{1-\left(\frac{2\,r}{2\,R}\right)^2}} = \sqrt{\frac{1.4 \times 32.2 \times 83.3}{1-\left(\frac{6.5}{18.75}\right)^2}} = 64.3 \text{ feet per second.}$$

Velocity of the water in 8-inch discharge,

$$V = \frac{241}{60 \times \left(\frac{\pi}{4}\right)\left(\frac{8}{12}\right)^2} = 11.5 \text{ feet per second.}$$

Assuming a suction pipe the same diameter as the discharge, the speed with 95 per cent volumetric efficiency would be $V_4 = \frac{11.5}{0.95} = 12.1$ feet per second. This is high, hence it would be well to select a 10-inch suction. A pump designed upon the basis of the given piping would probably not give over 65 per cent efficiency. Calculating the angles α and α_1 would furthermore show that an 8-inch diameter would not be as suitable for these con-

ditions, as a 10-inch pump, and that the speed should be increased to about 1000 revolutions in order to get good results.

Analyzing the angles, α and α_1 have to be taken so that they will make $V_1 > V_3$. We have, therefore,

$$V_3 = \frac{241.12^2}{60\frac{\pi}{4}(6.5^2 - 2.5^2)} = 20.5 \text{ feet per second.}$$

Taking tang α = 3, we have tang α_1 = tang $\alpha\left[\frac{V_5^2\eta_h}{gH} - 1\right]$

$$= 3 \times \left\{\frac{65^2 \times 0.68}{32.2 \times 83.3} - 1\right\} = 0.07 \times 3 = 0.21$$

and $$V_1 = \frac{V_5}{\text{tang } \alpha \times \text{tang } \alpha_1} = \frac{65}{3 + 0.21} = 20.3 \text{ feet per second.}$$

These angles, $\alpha = 72°$ and $\alpha_1 = 12°$, are too large for good results.

Example No. 3.—4-inch 12-stage.

Conditions.—150 gallons or 20 cubic feet per minute;
610 feet head, or 50.9 feet per stage;
1460 revolutions per minute.

Efficiency, volumetric = 90 per cent.
Total efficiency = 56 per cent.
Hydraulic efficiency = 62 per cent.

Diameter of impellers: hub, 2 inches; $2r = 3\frac{7}{8}$ inches inside; $2R = 9$ inches outside.

Speed for 9-inch impeller $= V_5 = \frac{\pi\left(\frac{9}{12}\right)1460}{60} = 57$ feet per second.

Speed to start under full head,

$$V_5 > \sqrt{\frac{1.4 \times 32.2 \times 50.9}{1 - \left(\frac{3.875}{9}\right)^2}} = 52 \text{ feet per second.}$$

Speed of water in suction and discharge, assuming the same diameters,

$$V = \frac{20.1}{60 \times \frac{\pi}{4} \times \left(\frac{4}{12}\right)^2} = 3.83 \text{ feet per second,}$$

$$V_4 = \frac{3.83}{0.9} = 4.26 \text{ feet per second.}$$

It would be advisable to make suction pipe 5 inches diameter in order to reduce velocity.

$$\text{Velocity } V_3 = \frac{20.1 \cdot 12^2}{60 \cdot \frac{\pi}{4}(3.875^2 - 2^2)} = 5.11 \text{ feet per second.}$$

Take tang α = 8, which will give

$$\text{tang } \alpha_1 = 8\left\{\frac{57^2 \times 0.62}{32.2 \times 50.9} - 1\right\} = 8 \times 0.23 = 1.84,$$

and $$V_1 = \frac{57}{8 + 1.84} = 5.8 \text{ feet per second.}$$

Thus $V_1 > V_3$, as it should be.

Angle α_2, speed V_2 = 5 feet. $\frac{5.8 + 5.1}{2 \times 1.1} = 5.$

$$\text{speed } V_6 = 57 \times \frac{3.875}{9} = 24.6 \text{ feet per second.}$$

$$\text{tang } \alpha_2 = \frac{24.6}{5} = 4.92.$$

Assuming 8 diffusion vanes and 12 impeller vanes at outlet, with thickness of diffusion vanes $\frac{11}{16}$ inch and impeller vanes $\frac{9}{32}$ inch, we have

$$W_1 = \frac{20}{60 \times V_1} \times 12 \div \left\{2 R\pi - 8\left(\frac{0.6875 \text{ inch}}{12}\right) - 12\left(\frac{0.28 \text{ inch}}{12}\right)\right\},$$

$$W_1 = \frac{0.333 \times 12}{5.8} \div \{2.36 - 8 \times 0.057 - 12 \times 0.023\},$$

$$W_1 = 0.69 \div \{2.36 - 0.456 - 0.276\},$$

$$W_1 = \frac{0.69}{1.63} = 0.42 \text{ inch.}$$

Allowing 10 per cent for losses, the width W_1 becomes 0.46 inch, or about $\frac{7}{16}$ inch.

Inside width W, therefore, may be similarly calculated, assuming 6 vanes at entrance $\frac{1}{4}$ inch thick.

$$W = \frac{0.333 \times 12}{5 \text{ or } V_2} \div \{3.14 \times 0.32 - 6 \times 0.02\},$$

$$W = 0.799 \div \{1 - 0.12\} = \frac{0.799}{0.88} = 0.896 \text{ inch.}$$

Allowing for 10 per cent, W = 0.985, or about 1 inch.

It should be noted that hydraulic efficiencies of 62 to 79 per cent should be used for volute type of pumps, about 62 to 70 per cent for small turbine pumps, and 70 to 85 per cent for large ones, the value varying with the type and design of the particular pumps considered.

A method which may be applied in a somewhat different manner is given below.

Radial velocity at outer circumference, V_1;
Radial velocity at inner circumference, V_2;
W_1 = width at outer circumference;
W = width at inner circumference;
f_1 = thickness of vane at outer circumference;
f_2 = thickness of vane at inner circumference;
k = coefficient of contraction = 0.90.

$$V_1 = ef \sin \alpha_3 = \frac{Q}{(2\pi R W_1 - n W_1 \operatorname{cosec} \alpha_3 f_1)\, k},$$

$$V_2 = \frac{Q}{(2\pi r W - n W \operatorname{cosec} \alpha_4 f_2)\, k}.$$

See Fig. 62.

The angle α_3 varies in practice from 15 to 30 degrees.

The width W_1 and W can be calculated from the following formulæ:

$$W_1 = \frac{Q}{V_1\left(2R\pi - \frac{zf}{\cos\alpha} - \frac{z_1 f_1}{\cos\alpha_1}\right)},$$

$$W = \frac{Q}{V_2\left(2r\pi - \frac{z_2 f_2}{\cos\alpha_2}\right)}.$$

Z = number of diffusion vanes.
Z_1 = number of impeller vanes at outlet.
Z_2 = number of impeller vanes at inlet.

These values of W should be increased about 10 per cent for internal leakage and losses.

Volumes should be calculated in cubic feet per second and dimensions in feet.

CHAPTER XIII.

SCREW OR PROPELLER PUMPS.

THE latest development of the centrifugal pump is its use combined as a unit with the steam turbine for circulating water at very low heads, and in moving large bodies of water under low velocities and at comparatively low heads. In this field, high speed causes the designer special difficulties in determining the proper inlet and outlet diameters of the impellers, and the length of the vanes; and frequently the path of the water in the impeller becomes too short. For such work the moving of the water in an axial direction is necessary, and a screw or propeller is particularly adapted to it. Screw pumps belong to the same class of velocity pumps as centrifugal. They consist of guide vanes at the inlet, with screw or propeller and guide vanes at the outlet. All are mounted in a cylinder or chamber. The velocity of the water, being parallel or axial, lends itself readily to operation at high speeds, with high efficiency for water at low heads. By arranging the screws or propellers in opposition to one another, all end or lateral thrust on the shaft is eliminated, and at the same time the capacity can be doubled without increase of speed. The angles of the stationary inlet and outlet vanes and blades of the screws are so arranged that the water enters axially and is given a radial motion, finally being discharged parallel to the axis in the vortex or volute chamber, thereby utilizing both the impulse and reactional forces.

In a centrifugal pump the liquid flows in a radial direction through the impeller and obtains its energy through the difference in circumferential speed at the inlet and outlet of the impeller. In a screw-propeller pump the liquid flows axially, there is no increase in circumferential speed, hence the liquid must obtain its pressure through some other cause.

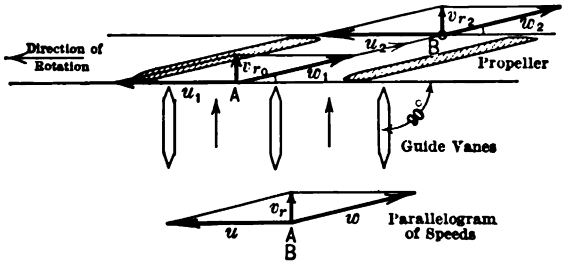


Fig. 69. Path of Water.

Suppose, first, that a screw propeller revolves in the water so that no shock occurs either at the inlet or outlet. The water flows as shown in diagrams at *A* and *B* (see Fig. 69).

u_1, circumferential speed at A, is equal to u_2, circumferential speed at B.

v_{r_0}, absolute inlet speed, equal to v_{r_1}, absolute outlet speed.

w_1, relative inlet speed, equal to w_2, relative outlet speed.

The diagrams at A and at B are similar. The liquid, therefore, has the same energy at A and B. During the flow of the liquid from A to B no energy has been put into the liquid and no pumping head can be produced. *A smooth, shockless flow through a screw propeller does not produce any pumping head.*

Slip. — A certain shock must, therefore, be produced in order to obtain pressure. The full theoretical capacity cannot pass through the propeller. This reduction in capacity is called *slip*. Instead of letting the water enter as per diagram $uv_{r_0}w$, we let it enter as per diagram ucw' (see Fig. 70).

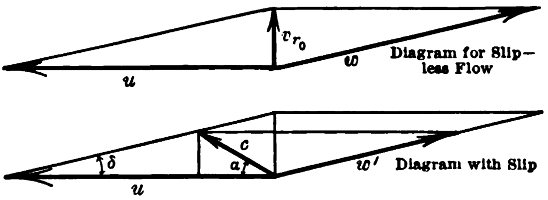


Fig. 70. Entrance Angles for Water.

Definition of Slip. —

Q_0 = capacity without slip, or the theoretical capacity.

Q = capacity with slip, or the actual capacity.

z = slip.

$$z = 1 - \frac{Q}{Q_0}.$$

$$\frac{Q}{Q_0} = 1 - z = \text{ratio between obtained capacity and maximum capacity.}$$

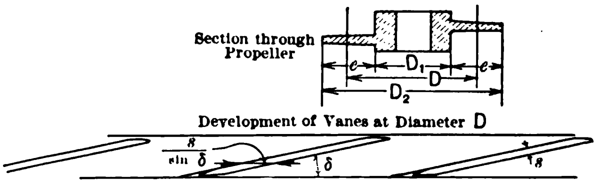


Fig. 71. Guide Vanes.

Assuming radial guide vanes we find as in Fig. 71,

$$Q_0 = A \cdot v_{r_0} = A \cdot u \cdot \operatorname{tang} \delta.$$

$$A = \text{radial area} = [D_2^2 - D_1^2]\frac{\pi}{4} - \frac{s}{\sin\delta} \cdot e \cdot z.$$

δ = pitch angle.

z = number of vanes.

$$u = \text{circumferential speed} = \frac{D \cdot \pi \cdot \text{r.p.m.}}{60}.$$

Assuming guide vanes entering at an angle ϵ in direction of rotation (see Fig. 72),

$$Q_0' = \text{theoretical maximum capacity;}$$

$$Q_0' = A \cdot u \cdot \operatorname{tang} \delta \frac{1}{\left(1 + \dfrac{\operatorname{tang} \delta}{\operatorname{tang} \epsilon}\right)}.$$

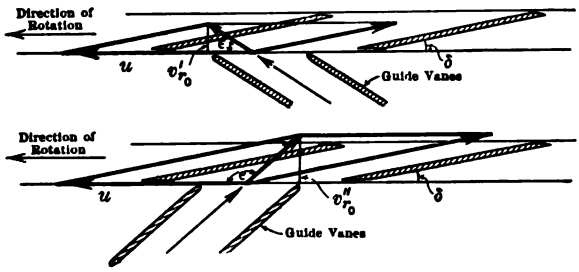


Fig. 72. Guide Vanes.

Assuming guide vanes entering at an angle ϵ, pointing opposite to direction of rotation,

$$Q_0'' = \text{theoretical maximum capacity;}$$

$$Q_0'' = A \cdot u \cdot \operatorname{tang} \delta \frac{1}{\left(1 + \dfrac{\operatorname{tang} \delta}{\operatorname{tang} \epsilon}\right)}.$$

For Q_0' the only difference in the equations is in the values of tang ϵ, which are positive and negative respectively for ϕ_0' and Q_0''.

$$Q_0'' > Q_0'.$$

If the inclination of the guide vanes is in the direction of rotation, the maximum capacity for slipless flow is smaller than for radial guide vanes. For guide vanes pointing against the direction of rotation, the maximum capacity is larger than for radial guide vanes:

$$Q_0' < Q_0 < Q_0''.$$

$$z = 1 - \frac{Q}{Q_0} \text{ for } \epsilon = 90°,$$

$$z = 1 - \frac{Q}{Q_0'} \text{ for } \epsilon < 90°,$$

$$z = 1 - \frac{Q}{Q_0''} \text{ for } \epsilon > 90°,$$

Q_0, Q_0', Q_0'' being figured by the above equations.

Law of Proportionality and Slip. — For water turbines and centrifugal pumps a law exists which can be expressed by the formula

$$\frac{N}{N_1} = \frac{Q}{Q_1} = \sqrt{\frac{H}{H_1}} \backsim \sqrt[3]{\frac{\text{H.P.}}{\text{H.P.}_1}},$$

where N = revolutions per minute;
Q = capacity;
H = total head;
H.P. = horse power.

This law implies that the capacity changes in direct proportion to the speed, the head changes in proportion to the square of the speed, and the horse power changes approximately as the cube of the speed. This law has been found to be correct within practical limits.

Propeller-pump tests show the interesting fact that this law of proportionality can also be applied to ship propellers.

For low speeds the tests show conformity in regard to speed, capacity, and head; for the higher speeds, however, and for the horse powers, some irregularities occur. The law of proportionality can be shown by the curves $\frac{Q}{N}$ = constant and $\frac{Q}{\sqrt{H}}$ = constant. See Fig. 73.

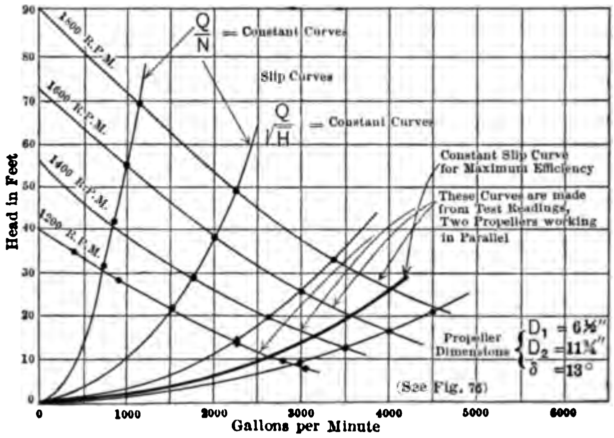


Fig. 73. Performance Curves.

It can be proved that if the law of proportionality is correct then the curves $\frac{Q}{N}$ = constant and $\frac{Q}{\sqrt{H}}$ = constant represent curves of constant slip.

$$\text{Slip} = z = 1 - \frac{Q}{Q_0}.$$

$$\frac{Q}{Q_0} = 1 - z \qquad Q = Q_0(1 - z)$$

$$\frac{Q}{N} = \frac{Q_0}{N}(1 - z)$$

$\frac{Q}{N}$ = constant. $\frac{Q_0}{N}$ = constant = law of proportionality.

If $\frac{Q}{N}$ = constant, then z = constant.

This result is of great importance, as it shows that *points of same slip must be points of similar efficiency.*

For changes of speed the best efficiency will occur at a constant slip. For differently designed propellers the best efficiency will occur at different slips. For a propeller of dimensions given in Fig. 73, the best efficiency, 66 per cent occurred at 1800 r.p.m. with 37 per cent slip. Fig. 74 shows the position of vanes, with free space or clearance and no overlap.

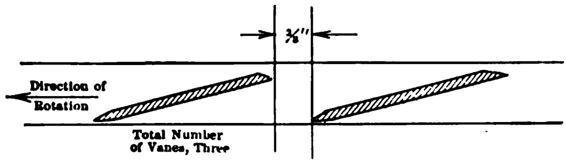


Fig. 74. Impeller Vane without Overlap.

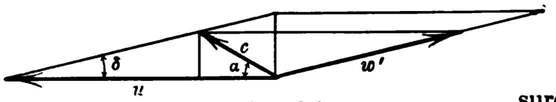


Fig. 75. Step Diagram.

Characteristic Curve.—Since slip is necessary in order to produce pressure, the relation between slip and pressure must be found.

This theory is based upon the assumption that all the shock produced by the slip is transformed into pressure, and that the pressure can be figured from this shock, provided a proper coefficient is applied.

On this basis we have the formula

$$H = z \cdot \frac{u^2}{g}(1 - \sin\alpha) \cdot y.*$$

z = slip;
u = average circumferential speed;
g = acceleration of gravity;
α = absolute angle formed by the direction of the inflowing liquid with the moving propeller vane;
y = coefficient.

See Fig. 75.

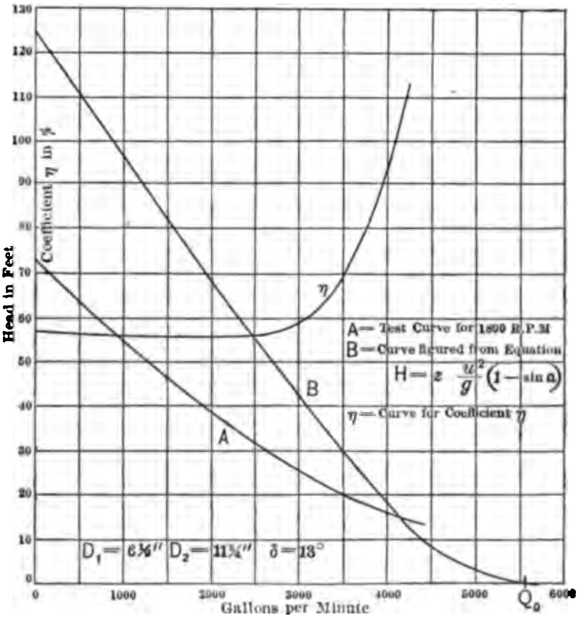


Fig. 76. Application of Equation Described.

While this equation does not give reliable results, it can, if properly applied, be used to give an approximation. Fig. 76 shows how this equation can be used.

* Hollander's formula.

The coefficient y stays nearly constant for a large portion of the curve; for the lower heads, however, it changes suddenly. It changes for various propellers and working conditions, and its value must therefore be determined by tests in each particular case.

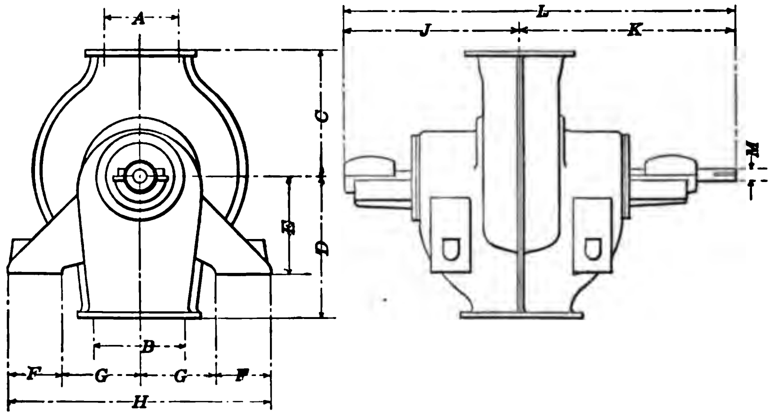


Fig. 77. General Arrangement of a Screw Pump.

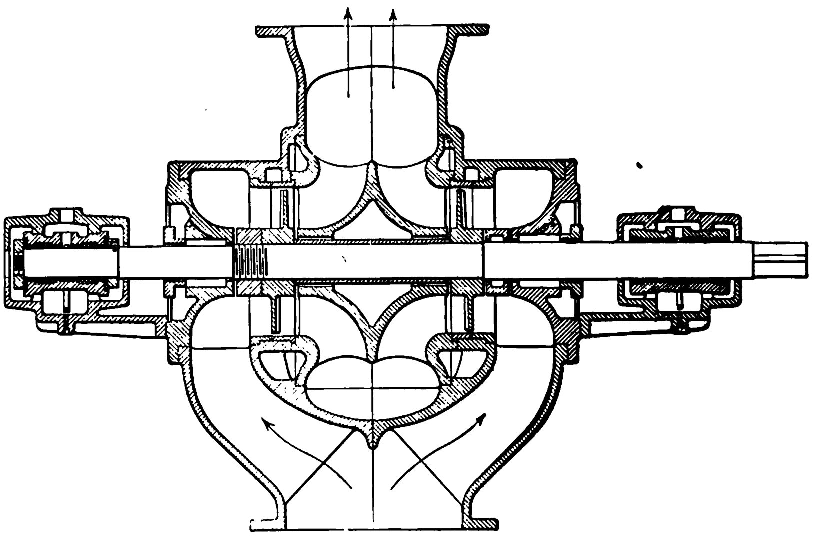


Fig. 78. Section of a Screw Pump.

A propeller pump is subject to the same general law of proportionality as water turbines and centrifugal pumps. A smooth water flow does not produce pressure: shock is a necessity. A high-speed propeller pump can, therefore, never have the same efficiency as a properly designed high-speed centrifugal pump.

The general arrangement of a screw pump is shown in Fig. 77, which gives approximate outside dimensions, and in Fig. 78, showing the pump in section. Fig. 79 is a section showing the combination of screw and volute centrifugal pump, adapted for a very low head. The inlet is trumpet-shaped to receive the current of water with the least loss. The impeller is

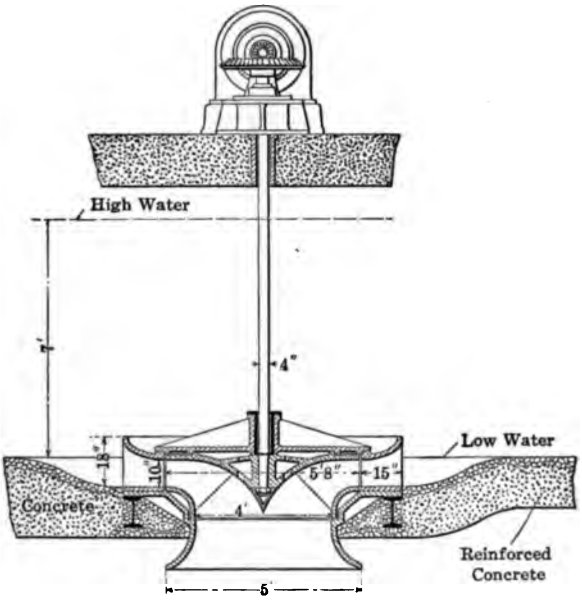


Fig. 79. Section Showing the Combination of Screw and Volute Centrifugal Pump.

formed by helicoidal vanes generated from the lowest part of the impeller and pitched backwards like a screw. From this form they gradually change into the regular shape of volute impeller vanes. The discharge passages can form a regular volute casing when water is to be delivered into a pipe or conduit. For canal work the open discharge casing is suitable in connection with the necessary sluice valves in the canal.

PART III.—APPLICATIONS AND USES.

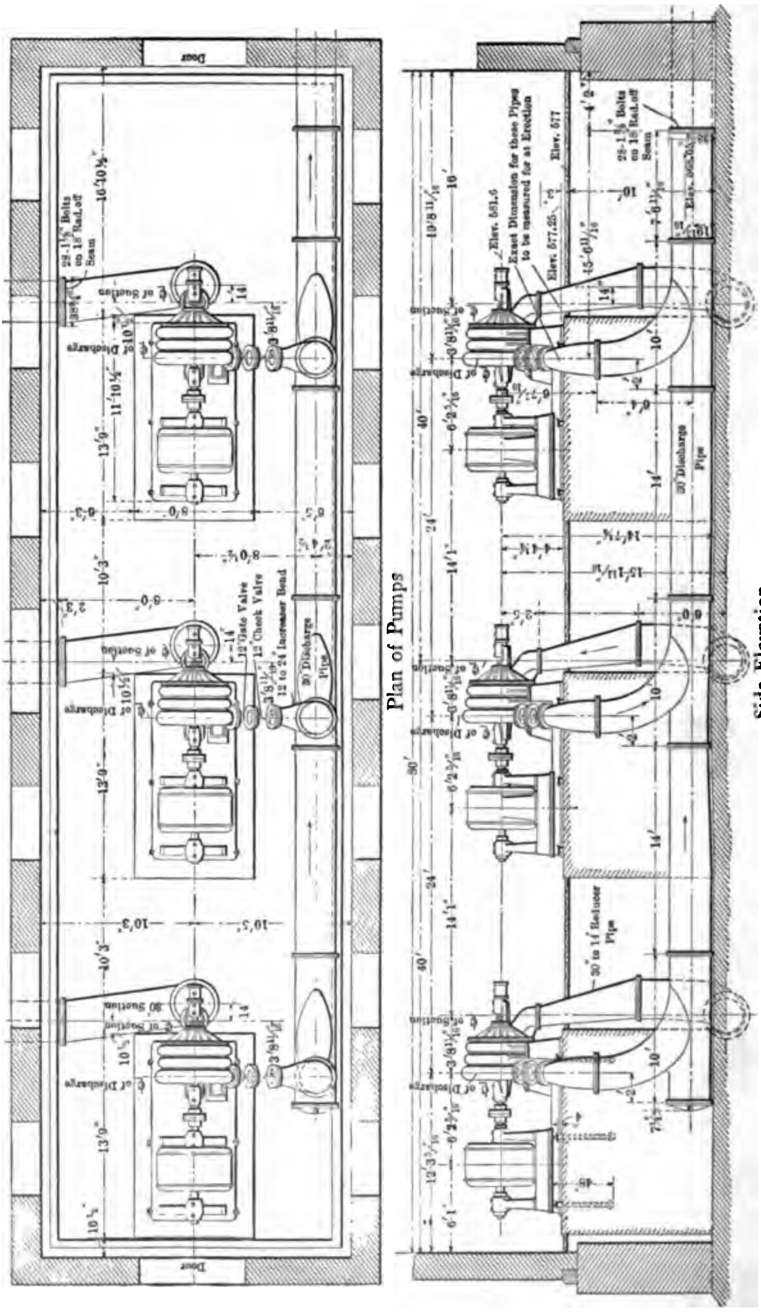
CHAPTER XIV.

GENERAL REMARKS.

WATERWORKS INSTALLATION.

Owing to its characteristics the centrifugal pump is better adapted to some engineering problems than to others. Its simplicity of construction, wide passageways, absence of valves, low first cost, comparatively light weight for its capacity and its adaptability for motordrives have given this type of pump a constantly widening field of application during the last fifteen years.

Fig. 80. Vertical, Self-contained, Two-million Gallon Turbine Pump.



Plan of Pumps

Side Elevation

Fig. 81. Plan and Side Elevation of Three 12-inch Three-stage Pumps.

In this and following sections are described the more general and important applications of the centrifugal pump up to the present time.

WATERWORKS.

Turbine pumps are now supplying water to municipalities like Buffalo, Lockport, Toronto, Montreal, Louisville, Minneapolis, and others, and furnish a constant and uninterrupted supply. The sizes range from small plants of a million gallons per day to those of twenty-five to thirty million gallons per day. In applying centrifugal pumps to service of this sort, the best practice calls for delivery to the top of standpipes or reservoirs. In this way both the capacity and pressure can be kept constant.

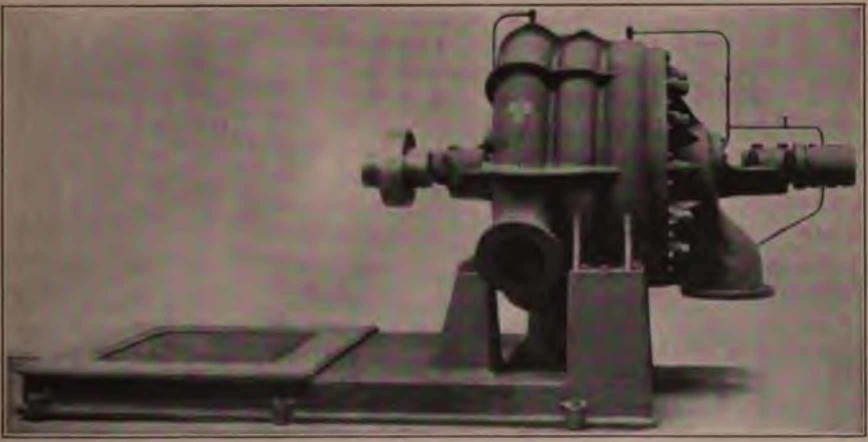


Fig. 82. Twelve-inch Three-stage Pump.

Fig. 80 illustrates a vertical, self-contained, two-million-gallon turbine pump installed at Athens, Ga., a type which takes up little floor space, and which has proved very satisfactory for small waterworks and manufacturing plants.

Figs. 81 and 82 illustrate the three 12-inch three-stage pumps of the waterworks plant of the city of Lockport, N. Y. Fig. 83 is the test curves of the motor.

The pumps are directly connected to 500-horse-power motors and are of the horizontal-shaft three-stage turbine type, with a single suction. The suction openings are 14 inches in diameter and fitted with special vapor openings. The main casing is cast iron of a tensile strength of 30,000 pounds, annular in form, and fitted with suitable supports to attach to base. The suction-head casting is of special design to facilitate the removal of internal parts without dismantling the pumps. The impeller is special bronze, of the inclosed type, and is arranged with balancing cham-

bers, which reduce the end thrust to a minimum. All impellers are perfectly balanced individually and when mounted together on the shaft. The discharge from each impeller is conducted through a set of guides or diffusion vanes, designed to transform the velocity into pressure with the least possible loss. These diffusion vanes are removable. The shafts are of nickel steel, ground and polished, and run in ring oil babbitted bearings.

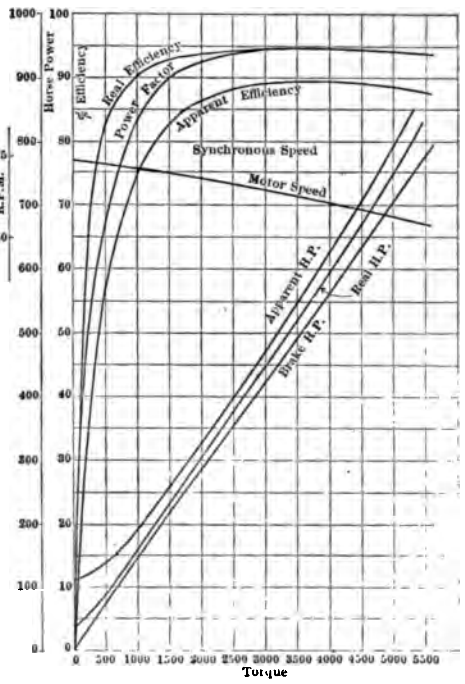


Fig. 83. Performance Curves of Motor Driving Pump, shown in Figure 83.

Stuffing boxes are fitted with water seals, consisting of lantern glands connected up with suitable piping, obviating all air leaks. All nuts are case-hardened, and the backs of the flanges spot-faced. The installation is a good sample of a modern turbine waterworks installation. Each pump has a capacity of 5,000,000 gallons of water in 24 hours, when operating at speeds given in the test. The motors were intended to run at 720 r.p.m., but actually run at 745 r.p.m. The pumps deliver, therefore, considerably more than the contract requirement, and maintain their efficiency. They were guaranteed to give 68 per cent efficiency, but actually give 82.5 per cent.

Water consumption in the city of Lockport is between 400,000 and 500,000 gallons per day and the water is delivered through 68,500 feet of 30-inch steel pipes from the Niagara River at Tonawanda, into a standpipe of limited capacity near Lockport.

The test was made with a constant level in the standpipe at Lockport, by throttling the discharge valves on the pumps until the Venturi meters showed that the pump was delivering at a five-million-gallon rate. The speed was 745 r.p.m. The gate valves were then opened and records taken for ratings of five and a half million, six million, seven million, and seven and three-quarter million gallons.

The following log of these tests is of interest because it shows the extraordinarily high efficiency and capacity of comparatively small pumps for waterworks service. A record was also kept of the pressures in the force main, the rate of pumping resulting, and the horse power of the motor as

indicated by the wattmeter on the gauge board. A complete record of the test is shown in the table.

Repeated tests on turbine pumps after years of service have shown that, when rightly constructed, there is little or no falling off in efficiency, a condition hardly met with in reciprocating pumps, where wear of valves is always present. In considering a waterworks or mill installation, original cost and maintenance should both be taken into account, as the turbine pump has many advantages over the plunger pump in lubrication, repairs, or replacement of operating parts, together with first cost, foundation, and building.

They should be designed for a fairly wide range of discharge, and usually for a constant head. Over-all efficiencies of from 68 to 72 per cent can be obtained as against 75 to 80 per cent for the reciprocating pumping engine. The latter still has an advantage in the cost of power, which, however, is offset by the lower cost of maintenance of turbine pumps, of their buildings, foundations, interest on investment, depreciation, and repairs.

TABLE NO. 1.—EFFICIENCY AT VARIOUS CAPACITIES AND PRESSURES OF 12-INCH WORTHINGTON 3-STAGE TURBINE PUMP NO. 3 AT LOCKPORT PUMPING STATION, NORTH TONAWANDA, N. Y.

Gallons discharged per 24 hours by Venturi meter in discharge line.	Total static pressure including suction lift.	Total dynamic pressure including suction lift outside of gate valve on pump.	Total friction loss through 69,000 feet of 30-inch pipe.	Theoretical friction loss through 69,000 feet of 30-inch pipe (Blackwell).	Total dynamic pressure including suction lift inside of gate valve on pump.	Pump horse power developed.	Motor horse power by wattmeter on switchboard.	Kilowatts by meter on switchboard.	Combined efficiency of pump and motor.	Efficiency of pump, assuming a motor efficiency of 92 per cent.	Revolutions per minute.	Useful pump horse power developed, based on pressure outside of gate valve on pump.	Combined efficiency of pump and motor based on useful pump horse power.	Efficiency of pump (motor efficiency of 92 per cent) based on useful pump horse power.	Remarks.
	Pounds	Pounds	Pounds	Pounds	Pounds				Per cent	Per cent			Per cent	Per cent	
5,000,000	80	93	13	12.9	160	323	442	330	73.1	79.5	745	188	42.5	46.2	Discharge gate throttled
5,500,000	80.5	97	16.5	15.6	154	342	453	338	75.6	82.3	745	216	47.7	51.9	Discharge gate throttled
6,000,000	80.5	101	20.5	18.7	147	356	470	350	75.8	82.5	742	245	52.2	56.9	Discharge gate throttled
7,000,000	81	106	25	25	127	358	497	370	72.2	78.5	742	300	60.4	65.8	Discharge gate throttled
7,750,000	81	108	27	31	108	338	520	388	65.0	70.6	742	338	65	70.6	Discharge gate full open

TESTS OF CENTRIFUGAL PUMPING ENGINES, MONTREAL.

The results of the tests of the engine-driven turbine pump at the low-level station of the Montreal waterworks have recently been made public. The plant consists of a Worthington three-stage turbine pump and a 750-horse-power triple-expansion steam engine as shown in Fig. 83A. The contract required that the unit should have a duty of 100,000,000 foot-pounds for each 1000 pounds of steam at 100 degrees superheat, allowance being made for all heat returned to the boilers.

Fig. 83A. — Worthington Three-stage Turbine, used in the Montreal Waterworks.

Test showed that the pump and engine reached a duty of 113,302,278 foot-pounds. The superheat was 166 degrees.

A second test gave a duty of 108,053,861 foot-pounds, at an average of 107 degrees superheat. This, with superheat at 100 degrees, would be equivalent to 107,588,268 foot-pounds.

A third test showed a total duty of 113,557,732 foot-pounds at 119.6 degrees superheat, which is equivalent to 110,151,000 foot-pounds at 100 degrees superheat.

The reason for the variation in the results is that the first test took place in the summer when the feed water was warm and second test took place in the fall when the water was very cold, and the colder the water the greater the value of the return heat. If allowance is made for the variation in superheat and feed water, the three tests are considered to check up so closely as to preclude errors.

TESTS OF CENTRIFUGAL PUMPING UNIT AT MONTREAL.

	Second Test.	Third Test.
Hours of test	24	13
Imperial gallons pumped	12,042,000	6,515,000
Average corrected head, feet	204.94	204.98
Work per hour, foot-pounds	1,027,772,000	1,027,267,000
Average corrected steam pressure, pounds	144.4	142.7
Average superheat at engine, degrees	107.4	119.6
Air pump discharge per hour, pounds	9,673	9,665
Duty per 1000 pounds steam condensed, foot-pounds	103,628,310	106,287,000
Duty per 1000 pounds steam supplied, corrected for 100 degrees superheat, and for heat returned to boiler, foot-pounds	107,588,268	110,151,000

The general appearance of the unit is shown in Fig. 83A. The surface condenser is located in the suction pipe and its air pump is driven by a belt from a pulley on the engine shaft near the coupling of the pump shaft. The total floor space occupied by the unit is about 23 by 40 feet.

In the second test, it was found impossible to keep the superheat at a uniform temperature on account of the existing conditions in the station. It varied considerably, falling below 100 degrees a number of times.

In the third test, the air pump discharge was weighed on calibrated scales, and the weighing was checked at frequent intervals throughout the trial. It was not possible to test the tightness of the condenser, but as any leakage there would go against the engine, and none was apparent, it was considered advisable not to consider this chance of error. The steam temperature was measured by a Hicks certified thermometer probably correct within $\frac{1}{2}$ degree Fahrenheit. The pump suction was measured by a mercury column. The water pumped from the heater to the boiler was measured by a meter which was tested. Feed water temperatures were taken with tested thermometers.

Owing to a burst main it was necessary to shut down the pump after running $13\frac{3}{4}$ hours. It was impossible to maintain the superheat at the specified amount, and the steam consumption actually measured was corrected in the following manner. The steam consumption per pump horse-power-hour and the average superheat for each hour were plotted on a diagram together with the guaranteed steam consumption of the engine. From this it appears that at 100 degrees superheat during the test the steam consumption would be 19.50 pounds per pump horse-power-hour as against 18.82 pounds at 120 degrees superheat, an increase of 3.6 per cent. From the curve based on the guarantee the increase per horse-power-hour was found to be from 12.85 to 13.25 pounds, an in-

crease of 3.1 per cent. It was, therefore, considered that it was fair to make an allowance of 3 per cent for the increase in the steam consumption when the superheat averaged 100 degrees instead of 119.6 degrees. The actual steam consumption as measured was, therefore, multiplied by 1.03 to allow for this.

CHAPTER XV.

IRRIGATION, DRAINAGE, AND SEWAGE.

THE work here is particularly well suited to centrifugal pumps, as the heads are low and the capacities large. Foreign substances such as sand, silt, and sewage matter must be handled, and as the centrifugal pump is

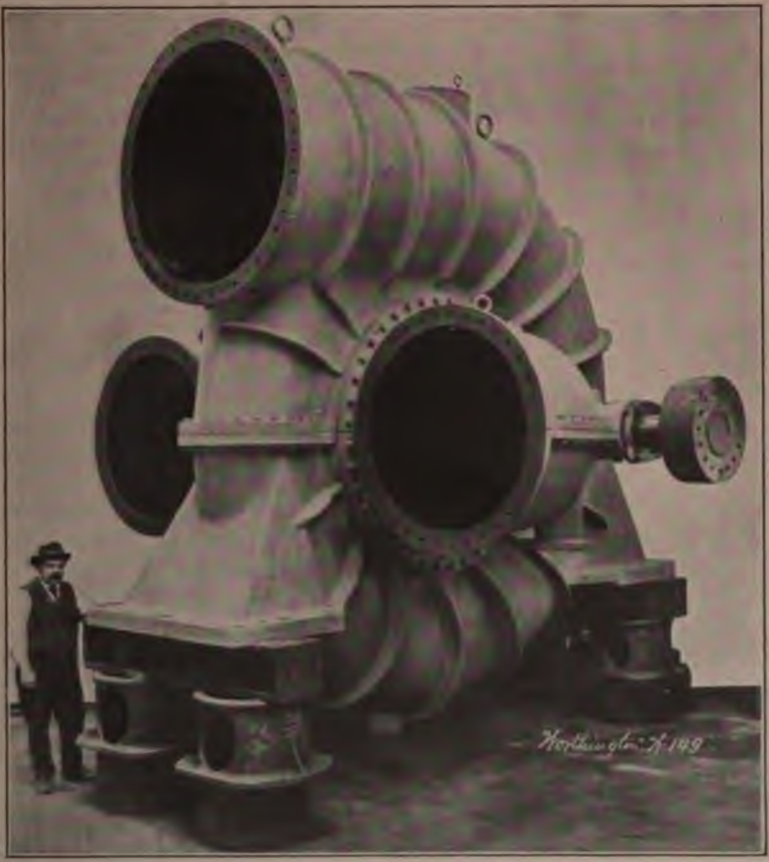


Fig. 84. 66-inch Irrigating Pump.

entirely free from valves and has large, easy passages, foreign substances pass through without choking or obstructing the pump. Water impregnated with acids, as is the case in sewage, does not tend to destroy them,

as in the reciprocating type, which offer more resistance and form adverse currents by their partitions, valve gratings, and valves.

The interior of a centrifugal pump can be readily coated with a non-corrosive enamel, covered with hot asphaltum and tar, which will form a solid enamel and prolong the life of ordinary cast iron to that of bronze.

In drainage work the prevailing lifts vary from 3 to 50 feet. It is well known that along the entire Mississippi River a great deal of land has been reclaimed, particularly near the bayous of Louisiana and Texas, where the rice crops now being raised do not depend upon the rainfall. Irrigation

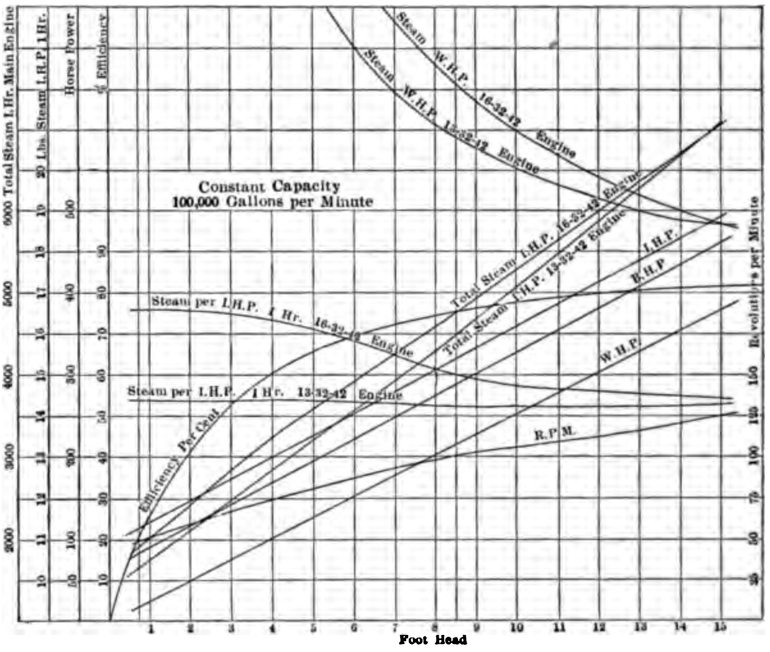


Fig. 85. Performance Curve of 66-inch Irrigating Pump.

through a series of canals now employed has revolutionized the rice industry. The amount of water required to flood these fields corresponds to an average depth of 2 feet or more for a season.

Egypt has again come to the front as a producing country, due to the introduction of modern centrifugal pumps, and is now covered with a series of canals supplied by large irrigating plants installed during the last five years, many of them of large size, ranging from 18 to 60 inches. The total amount of water these plants deliver is enormous.

Irrigation work is simply the reverse of drainage; in the former the water is lifted and distributed, and in the latter it is drained and discharged.

Fig. 84 illustrates a 66-inch irrigating pump for 100,000 gallons per minute, against a head of 8½ feet at 100 revolutions, and 125,000 gallons

against a head of about 20 feet at increased maximum speed of 150 revolutions; the former for 75 per cent efficiency and the latter 85 per cent.

Fig. 85 shows the performance curve of this pump at 100 revolutions.

Fig. 86 shows the installation of the Jennings Canal Company, Jennings, La., on the Bayou Nez Pique, a representative type. A description of this plant follows:

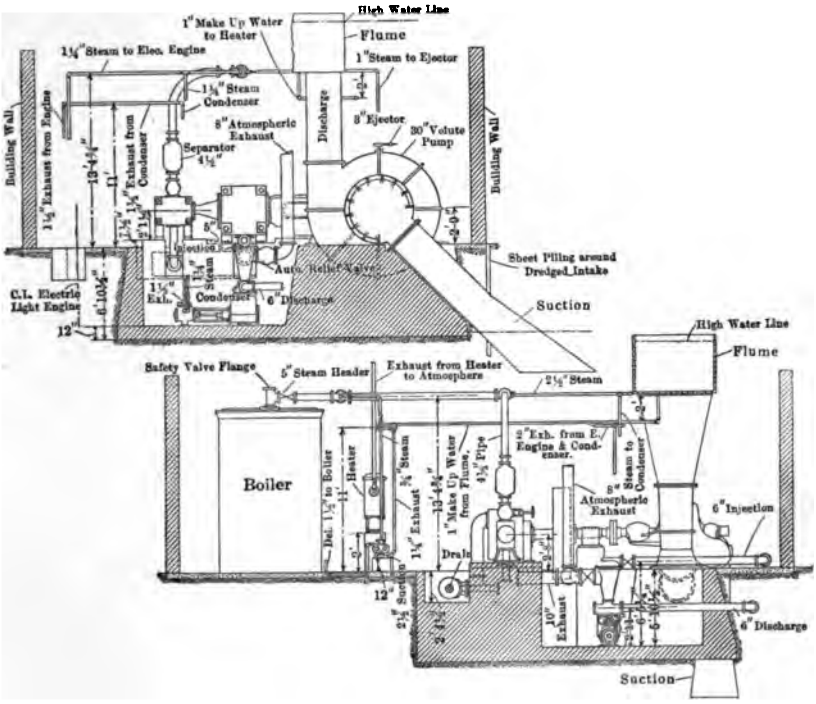


Fig. 86. Installation of a Worthington Centrifugal Pump, Plant of Jennings Canal Company, Jennings, La.

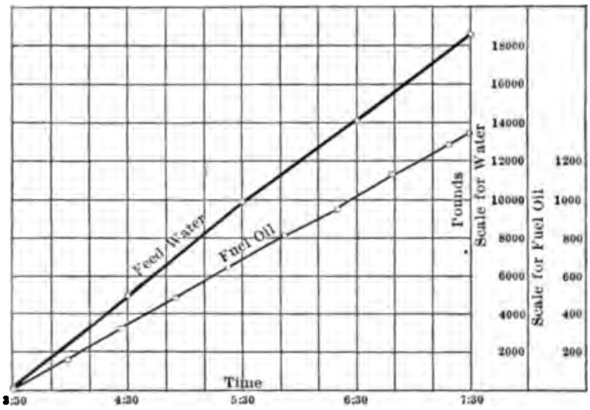
This is a Worthington centrifugal pump of the horizontal-shaft volute single-suction type; the suction pipe is 30 inches in diameter. The flume into which the pump discharges is built into the pumping plant and directly over the pump, so that it discharges the water upward through an expanding nozzle 30 inches in diameter; the top of the nozzle is rectangular in cross section and more than twice the area of the lower end, so that the velocity of water at discharge is greatly reduced.

The normal capacity of the pump is 22,000 gallons per minute, but can be increased to a much higher figure by an increase of speed. The pump is direct-connected to a Hamilton tandem compound condensing four-valve engine, size 12 inches and 24-by-24-inch stroke, provided with a shaft governor. The condenser is a Deane independent jet.

A Babcock & Wilcox water-tube boiler is installed which has 1550 square feet of heating surface. The steam pipe is short and 5 inches in diameter. The smokestack is 95 feet high and 32 inches in diameter. Feed water is supplied by a Worthington duplex piston pump $5\frac{1}{4}$ by $3\frac{1}{2}$ by 5 inches.

A Webster 175-horse-power heater raises the temperature of the feed water from 80° F. to nearly 170°. The fuel used is crude oil, costing 95 cents per barrel of 42 gallons, or 320 pounds, at the plant. The heat value of the oil is 18,922 B.t.u. per pound. It is pumped to the burner by means of a 3-by-2-by-3-inch duplex pump. The burner is a Peabody No. 1.

Some preliminary observations were made to find the economical speed, to adjust the engine valves, and to get the whole plant running under favorable conditions. The main test lasted for four hours, during which time all the observations were of great uniformity. As the boiler was of the cross-drum type with small water-storing capacity the length of time was sufficient for a test, which was satisfactory in every way.



The quantity of fuel oil measured at each reading was 160 pounds, and the time to consume this amount was never less than 28 minutes nor more than $29\frac{1}{2}$ minutes. Steam pressure was controlled mainly by means of the damper after the proper setting of the valves of the burner had been determined. The feed water was carefully weighed in barrels. At half-hour intervals indicator cards were taken, together with the general observations given in the log. Steam pressures were obtained by means of calibrated gauges. The amount of water pumped was measured in the flume by means of a Price current meter.

A calibrated vacuum gauge was attached to the suction pipe, from which the suction head was read. Although this pipe was enlarged at the lower end, there was some loss of head due to entrance loss and friction. Above the pump an opening was made in the discharge nozzle near the flange, where the diameter was 30 inches, and the discharge pressure read from a glass column attached by means of rubber tubing. The head on the pump and the difference of levels are given in the log.

This plant, when the size of the unit is considered, made a splendid showing. The amount of fuel oil per useful water horse power was 2.16 pounds.

SUMMARY OF RESULTS.

Test of Boiler, Engine, and Pump, Jennings Canal Company, *August* 10, 1907.

Average steam pressure gauge	145.4
Average temperature of water from heater, degrees Fahrenheit	169.5
Average temperature of water pumped	86.
Average temperature of flue	404.
Duration of boiler test, hours	4.
Total pounds of feed water	18,594.
Factor of evaporation	1.094
Total pounds of fuel oil	1344.
Amount of water in fuel oil, per cent	1.
Boiler horse power (basis 34.5 evap. from and at 212 degrees)	144.
Ratio of water evaporated to fuel oil, actual	13.52
Ratio of water evaporated to fuel oil from and at 212 degrees	14.8
Efficiency of boiler (18,922 B.t.u. per pound of fuel oil), per cent	75.5
Average rate of fuel oil per 24 hours, barrels of 320 pounds or 42 gallons	25.2
Duration of engine and pump tests, hours	4.
Average vacuum at condenser, inches of mercury	25.
Average revolutions per minute	187.7
Quality of steam at engine, per cent	97.8
Pounds of steam used by engine, vacuum and feed pumps and burner per I.H.P. hour	20.5
Average quantity of water pumped, gallons per minute	25,760.
Average head, suction level to discharge level, feet	24.04
Average head, including friction losses, feet	25.75
Average I.H.P.	221.4
Average useful water horse power	155.7
Efficiency of engine and pump, per cent	75.4
Efficiency of engine pump and pipe (including friction), per cent	70.3
Price of fuel oil, per barrel of 42 gallons or 320 pounds, dollars	.95
Barrels of fuel oil per hour	1.05
Pounds of fuel oil per minute	5.60
Heat value of fuel oil, by calorimeter, B.t.u	18,922.
Heat value of fuel oil per minute, B.t.u	106,000.
Heat equivalent to I.H.P. per minute, B.t.u	9370.
Heat equivalent to useful water horse power per minute	6590.
Ratio, heat equivalent of I.H.P. to heat value of oil	.0883
Ratio, heat equivalent of useful horse power to heat value of oil	.0622
Duty, millions of foot-pounds of useful work per million B.t.u. in fuel	48.3
Duty, millions of foot-pounds of useful work per 1,000 pounds of steam	67.7
Cost of fuel per hour, cents	100.
Cost of fuel per acre foot, cents	21.1
Cost of fuel per foot, acre foot, cents	.88
Cost of fuel to raise 1,000,000 gallons 1 foot, cents	2.68

TABLE NO. 2.—JENNINGS CANAL COMPANY, AUGUST 10, 1907.—*Concluded.*

Boiler test.														
Pressures.				Temperatures, F.°.						Pounds fuel oil.	Pounds feed water.	Draught, inches water.	Calorimeter.	
Boiler.	At throttle.	Receiver.	Vacuum.	Feed.	Injection.	Condenser discharge.	Oil.	Room.	Stack.				Temperature.	Pressure, feet, Hg.
142.3	....	12.0	25.0	....	....	....	....	....	....	....	....	....	....	....
142.3	....	13.7	24.9	....	....	....	....	....	....	....	....	....	....	....
144.3	144.0	11.6	25.0	....	86	....	....	103	394	....	....	.43	....	....
145.3	145.0	11.9	25.1	....	87	102	95	105	401	175	....	.43	....	....
145.3	145.0	11.6	24.9	166.6	86	....	91	104	405	340	4,828	.47	280	.49
145.3	145.0	11.7	24.9	....	86	109	90	103	....	515		.46		
145.3	145.0	11.6	25.1	173.0	86	....	85	101	408	680	9,888	.45	280	.51
146.3	146.0	11.6	25.1	....	85	108	....	98	....	845	14,054	....	....	....
146.3	146.0	11.6	25.1	160.0	85	....	....	99	414	1005	....	.46	280	.49
146.3	146.0	11.5	25.1	....	86	108	....	101	....	1190	....	....	....	....
144.3	144.0	11.4	25.1	179.2	86	....	....	104	....	1344	18,594	.46	....	....
145.4	....	....	....	169.5	86	....	....	....	404		....	....	280	.50

TABLE NO. 3.—JENNINGS CANAL COMPANY, August 10, 1907.

Engine and pump test.														
Time.	Revolutions per minute.	Indicated horse power.					Discharge.		Head.		Water horse power including friction.	Useful water horse power.	Efficiency.	
		High pressure.		Low pressure.		Total.	Gallons per minute.	Cubic feet per second.	At pump.	Difference in level.			Pump and engine.	Pump, engine, and piping.
		H	C	H	C									
9:30	196.0	69.25	66.65	67.30	53.90	257.1	27,500	61.20	26.25	24.31	182.0	169.7	70.8	65.6
11:00	177.6	56.85	54.80	42.70	40.85	195.2	20,900	46.50	24.80	24.02	130.7	126.5	67.0	64.9
3:30	185.6	66.20	52.80	60.20	44.80	224.0	24,900	55.50	25.30	24.00	159.2	151.0	71.1	67.4
4:00	187.8	68.20	54.15	56.80	45.00	224.2	25,750	57.25	25.30	24.12	164.2	156.5	73.1	69.8
4:30	187.6	67.45	53.00	56.20	43.85	220.5	25,700	57.15	25.79	24.03	167.0	155.7	75.7	70.6
5:00	187.0	67.10	53.40	55.50	44.60	220.6	25,700	57.15	25.69	24.10	166.4	156.1	75.4	70.8
5:30	188.0	67.40	53.80	56.30	44.60	222.1	25,900	57.50	25.92	24.02	168.9	156.5	76.0	70.5
6:00	188.0	67.20	53.10	56.80	44.55	221.7	25,700	57.15	25.81	23.96	167.2	155.2	75.5	70.0
6:30	188.0	67.65	53.60	56.30	44.55	222.1	25,600	56.90	25.85	24.00	166.8	154.8	75.1	69.7
7:00	187.6	66.65	54.15	55.10	44.50	220.4	25,500	56.75	25.95	24.09	166.9	154.8	75.7	70.2
7:30	187.6	65.50	53.60	55.40	45.00	219.5	25,800	57.40	25.90	23.99	168.5	156.1	76.8	71.1
......	*187.7	......	......	......	....	221.4	25,760	....	25.76	24.04	167.0	155.7	75.4	70.3
8:00	211	......	......	......	....	....	30,100	67.0	....	....	....	....	....	....

* These averages are for the last eight readings.

Fig. 87 shows the Dry Irrigating pumps, which were installed by the Madras Government. They comprise eight 39-inch volute pumps designed to run at 180 revolutions per. minute, direct-connected to vertical Diesel engines. Each pump will pump 26,000 gallons per minute against a head of 12 feet.

Fig. 88 shows the type of irrigating pumps in use in Colorado and places where water has to be elevated 100 feet or more. There are a number of these installations in operation for the Redlands Irrigating Company, the Orchard Construction Company, and several other companies which are rendering productive land which was formerly considered of little value.

The United States Government has introduced the policy of reclaiming land in the arid and semiarid regions that are located between the Missouri River and Rocky Mountains, and about 8000 canals and 3500 miles of ditches had been built up to 1909. The most important work has been

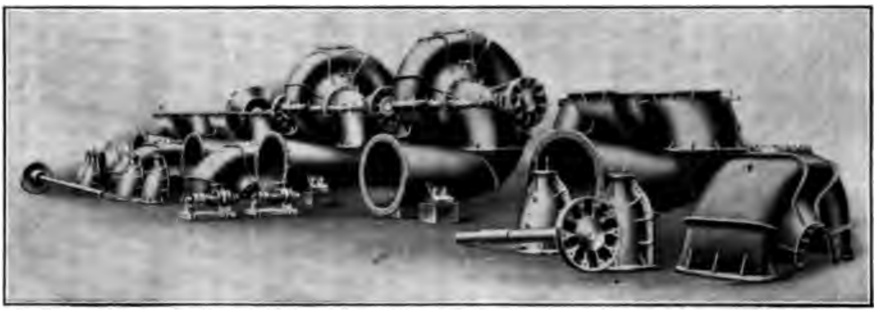


Fig. 87. Dry Irrigating Pump at Madras.

done in the North Platte district, which includes a part of Wyoming and Nebraska; the Shoshone district in Wyoming, with an elevation of 4000 to 5000 feet; the Salt River district in Arizona; the Huntley district of Montana, with an elevation of 3000 feet; the Wittestone district of North Dakota, where water is pumped from the Missouri River, and where the banks of the Missouri are so exposed to changes that floating pumping stations were made necessary; the Buford district, North Dakota; the Garden City district of Kansas, which uses an underground supply for irrigation. In all these stations, the pumping equipments are either of the motor-driven centrifugal or of the gas-engine-driven centrifugal type.

When a pumping station is required for occasional use only, the equipment should be simple, and economy is of secondary importance. For draining or irrigating large areas of land, where pumping is to be carried on uninterruptedly for months during the wet or dry season, there should be an economical installation, with carefully designed collecting canals, piping and outlets, so that water is not pumped even a few inches higher than necessary.

To secure an economical design, the pipes should be arranged to form a siphon, with the pump on the top of the siphon. The ends of the suction and delivery pipe form the ends of the siphon, which are determined by the lowest inner and outer levels reached under any working conditions. Suction pipes should be short, and plenty of room should be allowed around them so as to prevent currents or vortexes. Water should enter the suction pipe quietly and equally around the entire rim. The minimum diam-

Fig. 88. Type of Pump used in Colorado and Places where Water is to be elevated 100 feet or more.

eter of the suction well should be four times that of the pipe, and when properly designed the water may be pumped to within a foot of the rim of the pipe. The bottom or rim of the suction pipe should be of the bell form, known as vena contracta, which gives good results for low lifts.

It is also desirable to have the suction pipe gradually enlarge from the pump, so that the high velocities will be reached gradually. No screen should be fitted on the suction pipe itself, but one should be placed across the canal or well to prevent foreign matter from entering the pump.

The delivery pipes should be of gradually expanding section, so that the velocities at the point of discharge are reduced to from 4 to 2 feet per second, and should discharge into a still pool. This change of diameter in the pipe should take place in not less than 8 to 12 diameter lengths, otherwise the velocities will be changed too abruptly.

The remarks about irrigating and drainage pumps and the description of types apply in a great measure to pumps for sewage, of which there are many notable installations in such cities as Boston, Chicago, New Orleans, and Milwaukee, which give excellent results and good economy.

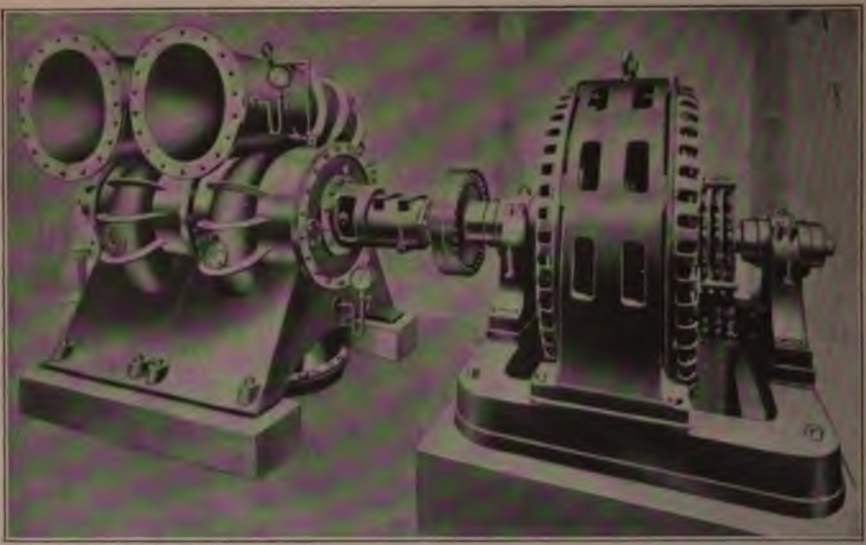


Fig. 89. Arrangement of Horizontal Sewage Pump installed by the Belfast Corporation, England, at the Greencastle Pumping Station.

Fig. 89 shows the arrangement of horizontal sewage pumps installed by the Belfast Corporation at the Greencastle Pumping Station, Belfast, England. In this case most of the pumping head is on the suction side, the pumps being automatically controlled by the suction level. Where vertical pumps are necessary, the type shown in Fig. 90 is used, which allows natural priming. Care should be taken in the design of pumps for unscreened sewage, as special impellers and large casings are necessary, and storm relief pumps must be so designed that they can handle considerable sand and grit.

Fig. 90. Arrangement of Vertical Pumps at the Greencastle Pumping Station, England.

CHAPTER XVI.

HYDRAULIC MINING AND DREDGING.

THE removal of earth in large quantities can be accomplished more quickly and cheaply by means of centrifugal pumps than with steam shovels. This is being done on the Pacific Coast in various places, both for mining and grading, and also at the phosphate mines in Florida.

To remove the loosened material special dredging pumps are used which are fitted with manganese impellers. The casings may be lined or unlined, depending upon the size of the pump and the nature of the work to be done. Such pumps are capable of handling a large amount of suspended solid matter, even up to 25 per cent of the water, and are designed so as to take care of stones 4 to 12 inches in diameter.

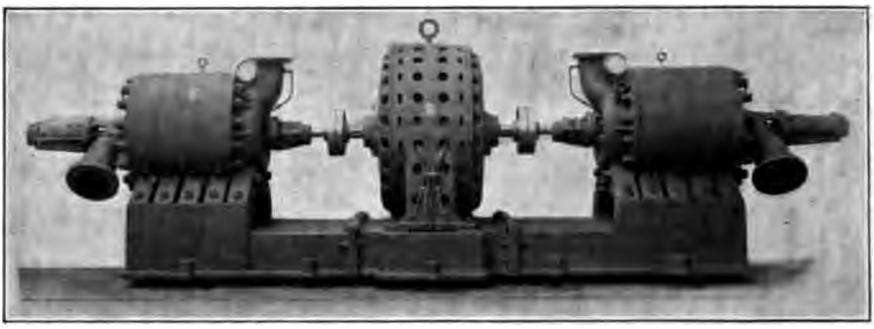


Fig. 91. 10-inch Ten-stage Hydraulic Mining Pump.

Fig. 91 shows one of the four 10-inch ten-stage pumps installed in Seattle for grading Jackson Street, used in removing two million cubic yards of earth.

The Panama Canal Commission have installed centrifugal pumping engines to remove 7,500,000 cubic yards of material, consisting of soft silt, earth, clay, and hard rock. Various attempts were made with dredges, but the final conclusion was to handle it by hydraulic methods. The material to be loosened and moved by the dredging pumps is dark loam, containing 15 per cent sand and gravel, and weighs 75 pounds per cubic foot.

Centrifugal pumps are frequently used for dredging and pumping mud from rivers and harbors. The cost of operating centrifugal dredging pumps compares favorably with ladder or bucket dredges, and in some cases is more economical. A modification of this pumping of suspended

material is found in wrecking work, where hulls are cleared of wheat and other substances which have no cohesion.

Centrifugal dredgers have been employed extensively by the United States Government for deepening rivers and harbors, and filling in and reclaiming land by pumping the dredged material through piping. Such work requires velocities in the pipes of about 10 feet for about 3 to 5 per cent of solid material, which consists of rock, clay, sand, shells, and mud. Discharge piping is made of either wood or metal, wood giving the better results in regard to wear and tear.

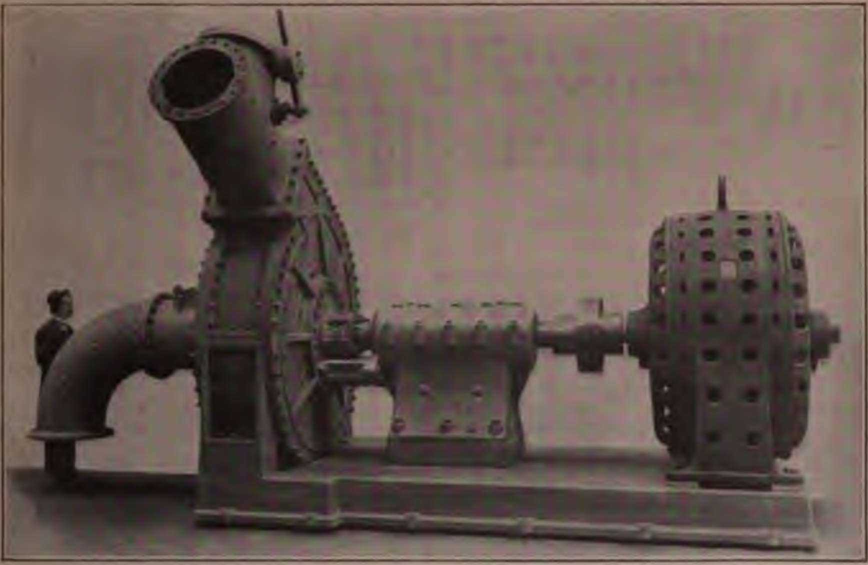


Fig. 92. Horizontal 20-inch Wilredging Pump. Horizontal Steel Lincel Type.

Fig. 92 shows the large dredging pumps installed at Panama by the Canal Commission. There are four of the horizontal-shaft type, each with a single 20-inch side suction and a 20-inch delivery, and are exceedingly heavy in construction. The runners or impellers are of manganese steel and of the inclosed type. The entire internal surface of the casing is covered with soft boiler plate, and the casings are of extra thickness to stand the heavy work. The suction openings are protected by removable cast-steel throats or rings. This construction makes one of the best arrangements for easy repair. The shafts are nickel steel.

Each pump is designed to deliver 10,000 gallons of water per minute, against a height of 60 feet, exclusive of pipe losses and suction heads of 10 feet through 1200 feet of pipe line. A suitable thrust bearing of marine type is provided to take care of the unbalanced pressure.

Each is capable of excavating and disposing of 300 cubic yards of solid matter per hour, using 10,000 gallons of water per minute, or about 10 cubic feet of water per cubic foot of material, which is loam with 15 per cent sand. With 3000 horse power available, the efficiency will reach approximately 60 per cent. On the basis of 10 cubic feet of water per cubic foot of material, and estimating the 85 per cent of dirt at 80 pounds per cubic foot, and the 15 per cent of sand at 110 pounds per cubic foot, we obtain 68 pounds and $16\frac{1}{2}$ pounds respectively, or a total of $84\frac{1}{2}$ pounds per cubic foot. Assuming a mixture of 40 per cent clay at 120 pounds per cubic foot, giving 40 pounds, 15 per cent sand at $16\frac{1}{2}$ pounds, and 45 per cent loam at 45 pounds, we have an average weight of $101\frac{1}{2}$ pounds for the material. Taking the weight at 110 pounds per cubic foot, and the volume at 300 cubic yards per hour, the material equals 891,000 pounds per hour, or 14,850 pounds per minute, plus 10,000 gallons or 84,042 pounds of water, making a total of 98,892 pounds of material to be pumped per minute by three pumps; the fourth being a "booster" pump. The total of 98,892 pounds divided by 10,000 gallons will give $9\frac{9}{10}$ pounds of weight per gallon, or $74\frac{1}{4}$ pounds per cubic foot of mixture. This gives 10 per cent of solid matter by volume and 15 per cent by weight.

Some of the mixtures to be carried weigh 90 pounds per cubic foot, which equals a weight of 12 pounds per gallon, or a total of 120,000 pounds per 10,000 gallons. Deducting the 84,042 pounds of water leaves 25,958 pounds of solid material, or about 21 per cent by weight.

These figures give an idea of the hard service under which these pumps operate. Each is driven by a 655-horse-power motor of the two-bearing type, running at 480 revolutions and fitted with autostarters and autotransformers. There are also four motors operating the vacuum pumps of the priming equipment. These vacuum pumps are of the single horizontal type operated by a 5-horse-power motor. There is also a complete set of transformers and marine switchboard for starting automatically, fitted with the necessary wattmeters, oil switches, and overload trip arrangement. There are all the accessories for starting the motor on the compensator and cutting it out when the switches are in running position. The current for operating the oil switches is 25 cycles, stepped down from 2080 volts to 110 volts. The motor connections are brought to a box near the base at the side of the motor. All of the leads are insulated and bushed to insure safety in case of accidental flooding.

The six hydraulic giants or monitors are of the latest type used in mining operations in California and other parts of the West. Each weighs 1500 pounds and consists of a base for attachment to a 16-inch gate valve at the terminus of the pipe line, a horizontal and vertical joint, and a long conical reducing piece. The frictional resistance is decreased by a ball bearing, and a weighted lever is attached to control the direction of the

jet. A deflecting nozzle is fitted to the discharge end of the giant, which permits deflections through a small angle without changing the position of the main body. The tapering piece of the giant is fitted on the inner side with two sets of guide vanes which prevent a scattering or rotary motion of the water after it has issued from the nozzle. The nozzles used vary from 4 to 6 inches in diameter, according to the character of the material in which they are working, and at full head the water coming through them exerts a pressure of 130 pounds to the square inch, the equivalent of a ton and one-half of pressure against a bank 100 feet away within range of the deflectors. As it is expected that the positions of the monitors will be shifted frequently, their bases are of temporary construction. The areas excavated and filled by the giants are 8 feet above mean tide and the average depth to be excavated is 45 feet. This is accomplished by washing down the material in sluices which carry the water containing earth in suspension to the sump where the barge or dredging pumps are at work. When it becomes necessary to move a barge, the giants cut a new sump with a channel leading into it through which one or more units of the fleet is floated. The banks are excavated as nearly perpendicularly as possible, in order that benches may be cut in them and the banks undermined so as to cause the material to fall by gravity.

CHAPTER XVII.

MINING WORK.

THE turbine pump has been extensively used for removing water from mines, for station or drainage service, and for sinking.

Fig. 93 shows an 8-inch eight-stage pump for a lift of 1250 feet. Several of these pumps are in successful operation handling large quantities of

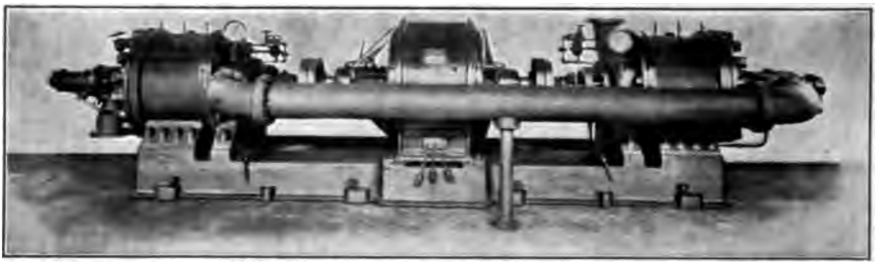


Fig. 93. 8-inch Eight-stage Pump. Deep Mining Pumps.

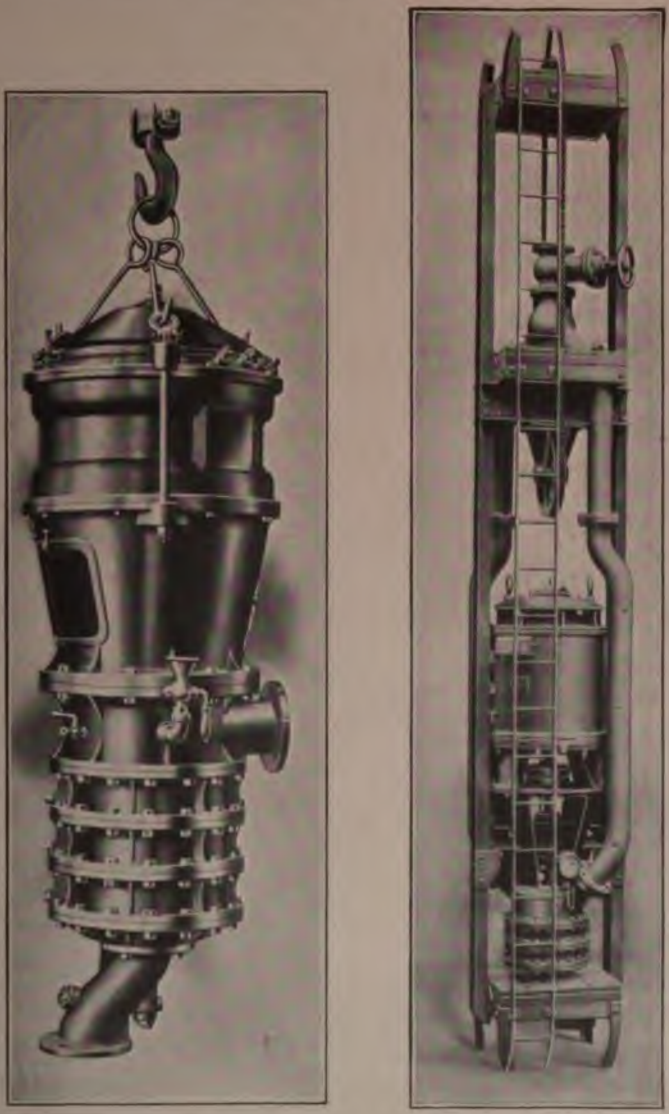
water, each impeller operating against as high as 157 feet, the efficiency being 75 per cent.

In a turbine type of pump it is practicable to have single lifts as high as 1600 to 1800 feet. In reciprocating pumps the lifts rarely exceed 1000 to 1200 feet. Several large pumps handling 5000 to 6000 gallons per minute, against 500 feet lift, have been in successful operation. For acid water the working parts are made of special acid-resisting bronze, containing copper, tin, and lead.

The use of electric sinking pumps is becoming general, particularly for emptying mines that are flooded. In sinking work a simple suction pump is provided, so that the suction pipe cannot be uncovered. This allows the pump to operate continuously with an ample supply of water.

Figs. 94 and 95 illustrate sinking pumps for deep working. The use of this type eliminates all steam and exhaust pipes. It is to be noted that a sinking pump should be proportional for the final head against which it has to pump, and special precautions must, therefore, be exercised in its design. The pumps are either self-supporting or hung in frames. This freedom from shocks, particularly in freely suspended pumps, allows high velocities of water, thereby reducing the size of pipe column.

By means of specially designed gate valves in the discharge, the quantity of water delivered can be regulated to the flow into the pump, and by a special design of suction chambers the accumulation of air can be prevented.



Figs. 94 and 95. Sinking Pumps for Deep Working.

The advantage of a centrifugal sinking pump lies in its compactness and light weight, with the consequent facility in handling and transporting in the shafts.

In introducing the turbine pump for mining purposes, considerable prejudice will have to be overcome, on account of the unsatisfactory results which have been due to wrong selection of pumps, and to the fact that manufacturers have not studied the requirements properly. The voltage may vary from outside causes, and a motor-driven turbine pump should be so installed that it will be reliable under all conditions.

If the speed of the motor and pump under the fluctuations of the line voltage and variation of capacity, and also the power of the motor against a constant head, be carefully considered, and the pump designed accordingly, there is no reason why it should not be used satisfactorily. The efficiencies obtainable are dependent upon the capacities for the heads under which the pump is to be installed, and attention should be paid to that point if good results are to be expected. With the proper relation between capacity and head, efficiencies equal to reciprocating power pumps may be expected, and with provisions for taking care of the fluctuations in the line, centrifugal pumps should meet with no difficulties.

CHAPTER XVIII.

POWER-STATION WORK.

BOILER FEEDING.

It is only lately that this promising field for the multistage turbine pump has been invaded. The turbine pump is well adapted to medium and large power-station work, where it will furnish an uninterrupted and continuous flow of water, free from shocks and water hammers, obviates the otherwise necessary air chambers, relief valves, and has a distinct advantage over the reciprocating pump in point of lubrication and attendance. The life of both piping and pumps will be prolonged, and the plant will be increased in efficiency. It is advisable in large power stations, however, to have a stand-by pump of the reciprocating type to be used in connection with the turbine pumps. There is no possibility of building up undue pressure by closing the discharge or feed valves. No trouble will come from steam-pressure regulators and safety valves if properly installed, but judgment must be used in relation to the drop in steam pressure, to low water, and peak loads; and a reciprocating pump must be kept for emergency and to take care of variations in capacity and irregularities.

Centrifugal pumps, if motor-driven under constant speed, should be designed for a considerable range of delivery that increase of pressure will be taken care of, with corresponding capacity. In a steam-turbine-driven pump this can be accomplished by increased speed. A motor-driven pump can also be arranged for variation in speed if desired. There is no fear of accident with a turbine pump, whether motor- or steam-turbine-driven.

Quite a number of these pumps are installed in large power stations like the Commonwealth station at Chicago, the New York Central power station at New York, the Interboro power station, and others, and abroad in the Edison Milano Univeraria Valdarno, in Turin, Milan, Florence, Birmingham, Amsterdam, and other places.

Fig. 96 illustrates a steam-turbine boiler-feed pump for 250 pounds pressure. Fig. 97 shows a steam-turbine-driven pump for 350 pounds pressure. Fig. 98 illustrates a motor-driven boiler-feed pump for 250 pounds pressure.

In boiler-feed pumps it is particularly desirable to have perfectly balanced conditions and free, uninterrupted flow to impellers, with velocities as low

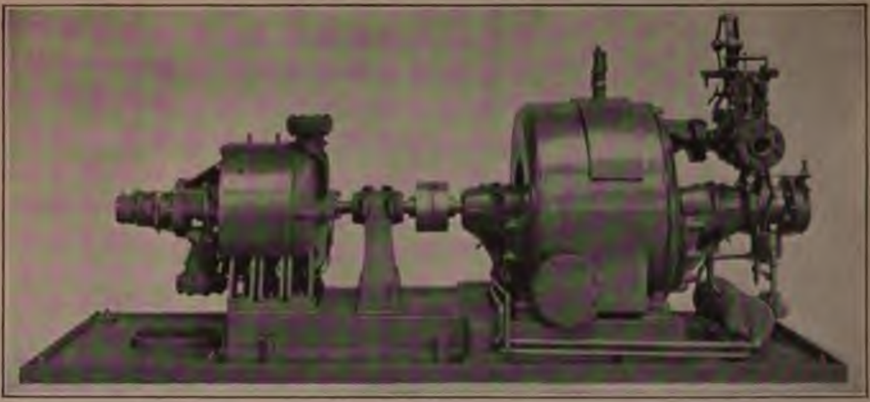


Fig. 96. Steam-turbine Boiler-feed Pump.

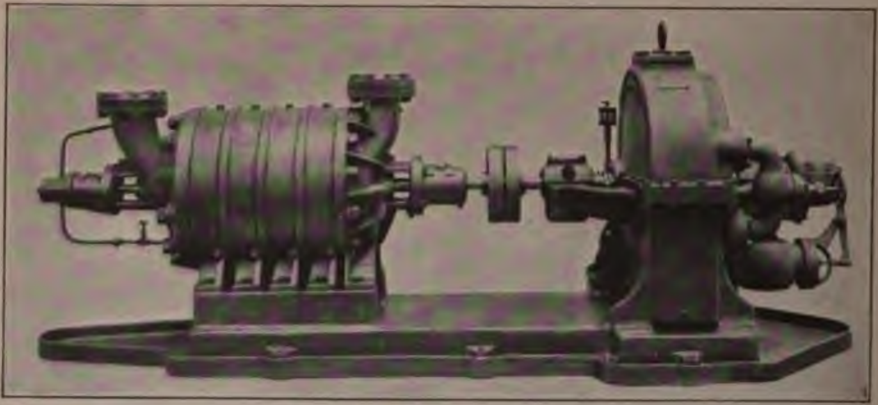


Fig. 97. Steam-turbine-driven Boiler-feed Fump.

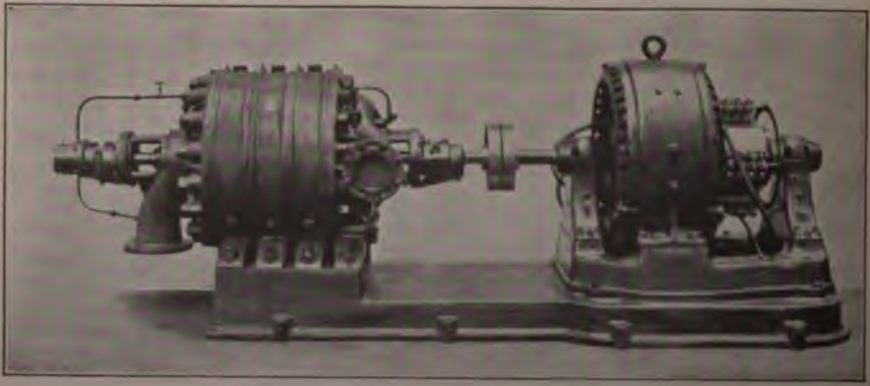


Fig. 98. Motor-driven Boiler-feed Pump.

as possible. An excellent design is shown in Fig. 99, where this is accomplished by admitting the water simultaneously to two impellers turned back to back, which discharge it into a central double-entrance impeller.

Good engineering requires the use of the simplest, most durable, and most economical apparatus. Since this type of pump has been introduced there is a tendency to do away with cumbersome and costly power pumps and steam reciprocating pumps, as the turbine pumps now installed show

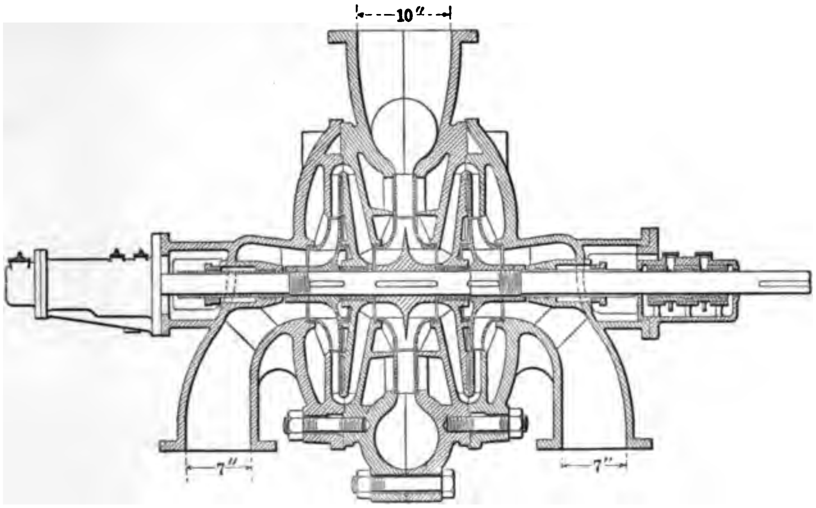


Fig. 99. Plan of Boiler-feed Pump with Opposed Balanced Impellers.

remarkable results for high speed, low cost of maintenance, reliability and smoothness of operation, economy in space, low cost of installation, and easy attendance.

CIRCULATION OF WATER.

In order to obtain circulating water for condenser plants, it is usual to employ a centrifugal pump on account of its high efficiency under the conditions which call for large volumes of water under low heads. Several designs are used, either engine-, motor-, or steam-turbine-driven, having a single or double suction, depending upon the conditions.

Condenser overloads should be taken into account, and the capacity of the pumps should be great enough to take care of the extra work. The connections to the pump from the condenser should be as short as possible, with no air pockets. Means should be devised for priming the pump, or a small connection may be made to the vacuum pump.

In engine-driven pumps the regular type of volute design of pump chamber and impeller can be easily applied, as the speed is normal. The siphon arrangement as used in irrigating work is most desirable. By specially

designed high-speed impellers this type can be used with motor or steam turbine drives within certain limits (see Fig. 100). But to meet the higher speeds of steam turbines and to handle large quantities of water under small heads, usually only friction heads, a new birotor or trirotor pump has been placed on the market. These pumps, described in Chapter XXIX, can be made in sizes and speeds to run with the average steam turbine.

In large condenser installations where all the auxiliaries are to be driven by steam turbines, a new type of centrifugal vacuum pump of the jet style

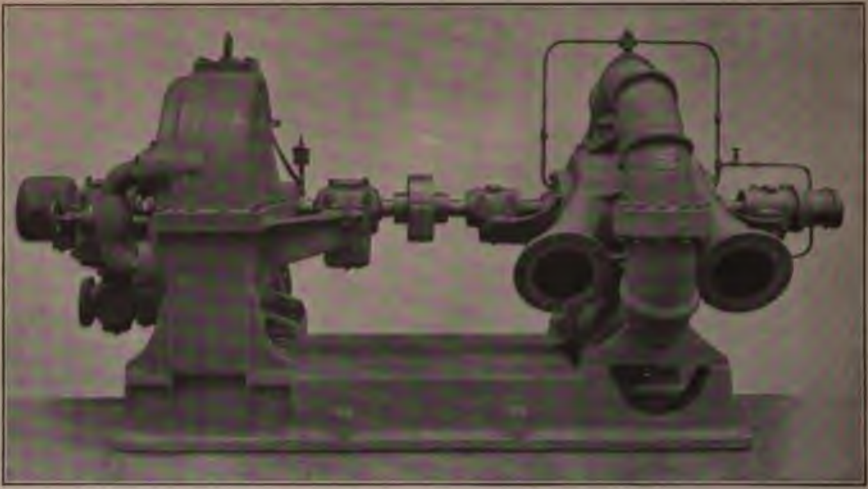


Fig. 100. Steam-turbine-driven Pump with High-speed Impellers.

has been developed. Several of these are being built, arranged on the same shaft with the circulating and hot-well pumps, making a very compact unit, the three auxiliaries being driven by one steam turbine.

HOT-WELL PUMPS.

Closely allied to the boiler-feed pump is the hot-well pump, designed for both steam-turbine and motor drive. It is rarely made more than two-stage, and has an additional vapor opening on the suction entrance. These pumps withdraw the water of condensation from the condenser. They may be either of the vertical or horizontal type, as most suitable to the particular installation.

Fig. 101 shows a pump inclosed in the hot well, making a very compact arrangement. Fig. 102 shows the regular arrangement.

The turbine hot-well pump has practically replaced the reciprocating type in all modern power stations, and has the advantage of being automatic in action, taking care of all the variations in the loads without any

attention, and is always in condition to operate without priming. These pumps are direct-connected to either a steam turbine or motor. It is usual

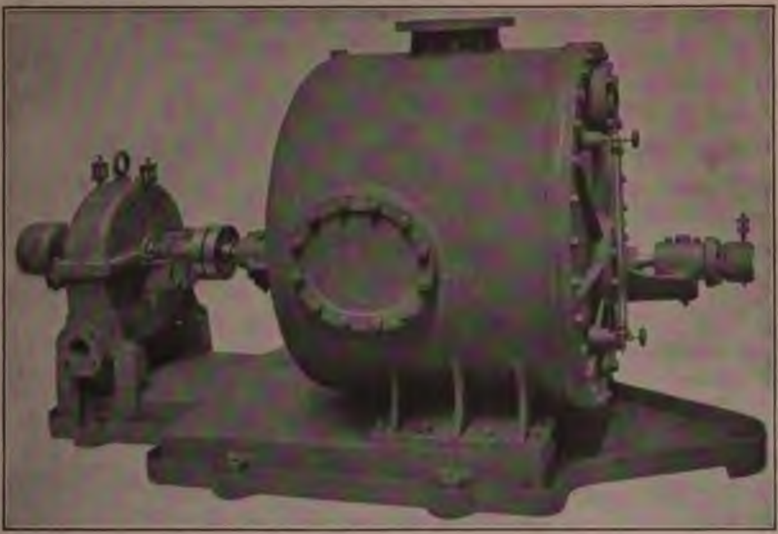


Fig. 101. Hot Well Pump Inclosed in Special Well.

to locate them three or four feet below the water line in condenser in order to obtain a good head on the pump. A type of the vertical pump installed

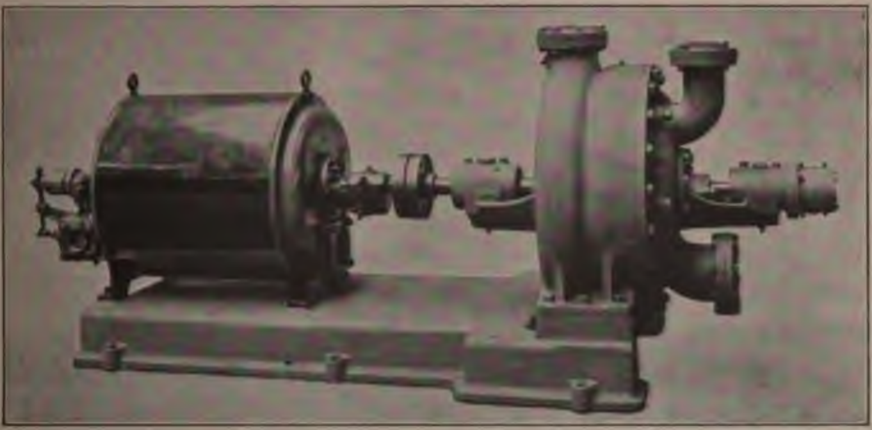


Fig. 102. Two-stage Hot Well Pump.

aboard ship is shown in Fig. 103, illustrating the compactness and lightness required in such installations.

CENTRIFUGAL-JET CONDENSERS.

The development of steam turbines requiring high vacuum has obliged the designers of centrifugal pumps to provide apparatus which will give a vacuum within one-half inch of the absolute vacuum. Barometric condensers have been used more or less for this work, but, owing to the extremely long piping with consequent air leaks, they have given way to the centrifugal jet.

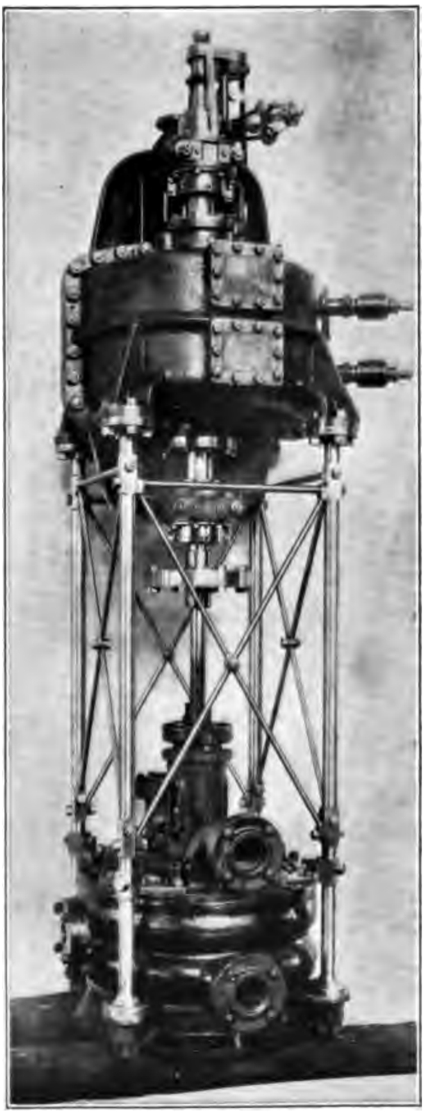


Fig. 103. Steam-turbine-driven Vertical Marine Hot Well Pump.

Fig. 104 illustrates an engine-driven unit which can also be driven by a steam turbine or motor. It contains some new features in the design of the working parts of the pump, and in the condenser, and may be built either on the two-stage or single-stage principle of the pump, depending upon the conditions. The principle involved is simple and offers the designer full range for his ability for getting good results with few working parts. The water is taken from the bottom of the condenser and led to the center of the impeller, when the condenser is alongside of the pump. When the condenser is on top of the pump, the water is guided to each side of the impeller. The removal of the non-condensable vapors is accomplished either by a rotative dry-vacuum or by a centrifugal vacuum pump. From the illustration it can be seen that compactness and space have been considered of first importance.

Figs. 105 and 106 show the arrangement for vertical and horizontal jet condensers, with trimotor and birotor pumps, driven by steam turbine, motor, or engine. Figs. 107 and 108 show a centrifugal air-jet condenser in which cold water for the condensation of the steam is forced through a

Fig. 104. Engine-driven Pump Unit.

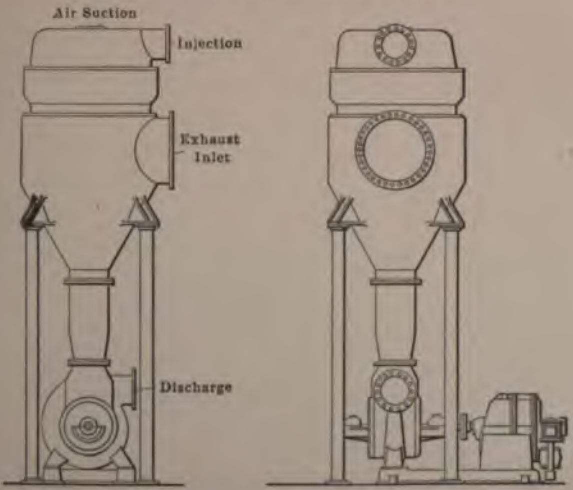


Fig. 105. Arrangement for Vertical Jet Condensers.

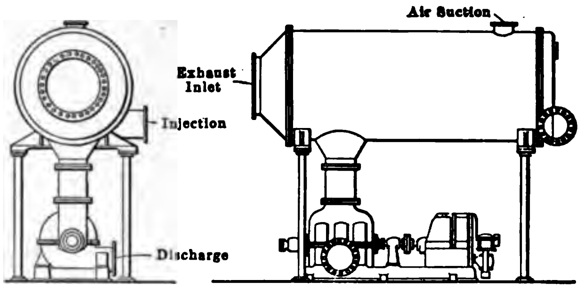


Fig. 106. Arrangement for Horizontal Jet Condenser.

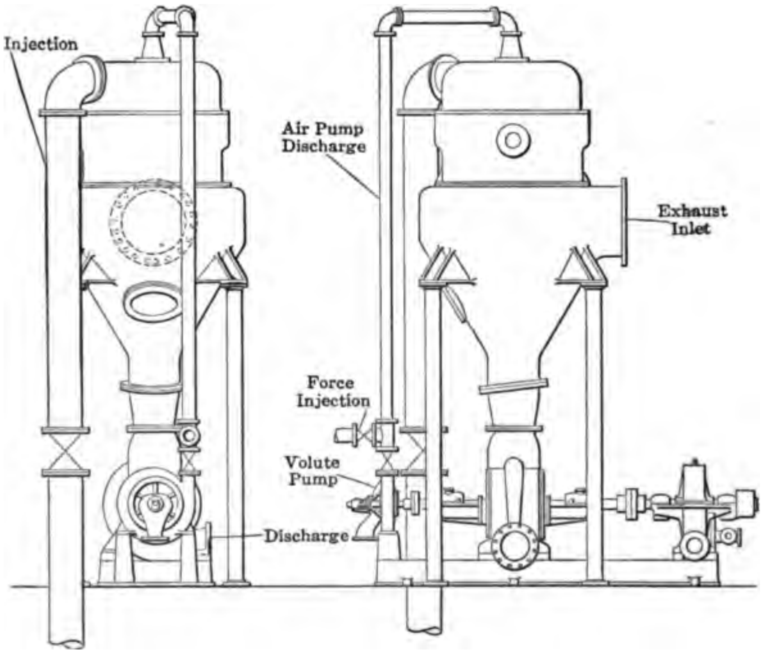


Fig. 107. Centrifugal Water Jet Condenser with Centrifugal Air Exhausting Pump.

jet nozzle, absorbing all the vapor contained. This method of extracting the air dispenses with the usual dry-vacuum pump. The water and condensed vapors are removed by a circulating pump in such a manner that

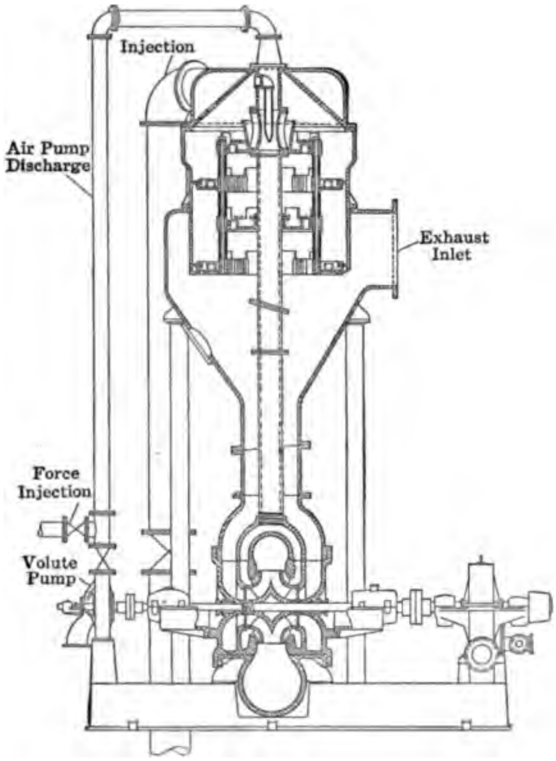


Fig. 108. Section of a Centrifugal Water Jet Condenser with Centrifugal Air Exhausting Pump.

the air is drawn off by the outside impeller vanes of the circulating pump. This construction allows direct pumping, and obviates a separate air or vacuum pump, and so produces a compact unit of low cost, high efficiency, and minimum expense for operation and maintenance.

CHAPTER XIX.

DOCKS.

One of the most extensive and important uses of the centrifugal pump is its application to pumping out docks, and it is here that the largest individual units, ranging from 36- to 72-inch discharge opening, are used.

This service calls for the handling of large volumes of water in the least possible time under heads varying from 0 to approximately 48 feet for the drainage pumps, and 0 to 40 feet for the main pumps. Great difficulties are met, as alternating current is largely used, necessitating constant speed and constant horse power over the entire range of capacity and head. A great many pontoon and floating dry-docks have been equipped with this type of pump, and have shown that the time taken to pump out docks may be lowered considerably, and that the electric power required may be reduced about 40 per cent below that which was used by the old-style pumps.

It is to be noted that in a dry-dock the sectional area decreases as the lift increases, and that the most economical heads will be found from about one foot above keel block to the bottom of dock.

The largest volumes are pumped at the smallest heads, and vice versa, therefore the average efficiency is very important, and in considering this average efficiency note should be made between what levels it is to be considered.

The capacity is usually judged on the basis of an average between zero head at mean high-water elevation and an elevation 1 foot above the top of the keel blocks, which is usually from 4 to 6 feet above the bottom of the dock. The keel block having the greatest elevation is the one usually selected as the pumping point, correction being made for tidal variations. In this way the results are reduced to a condition of constant mean high-water level outside the caisson during the operation of pumping.

Dock pumps are of both the vertical and horizontal type, the latter being the more common. The efficiencies in dry-dock main pumps should be judged on the average operating conditions given, and should be the ratio of the work done in pumping against the head to the power input of the motor measured at the motor terminal. The head for the main pump is usually taken from zero at mean high-water level, encountered when the level of the water in the dock is 1 foot above the keel blocks.

The efficiency of a well-designed dock equipment will reach about 42 per cent rated on the above basis, and about 45 per cent from the bottom of

the dock, the pumping unit being charged with all frictional losses in piping, valves, etc., with a motor efficiency of 92 per cent and from 85 to 90 per cent efficiency for the pumps themselves. Since all such pumps are required to run only for short periods and at long intervals, efficiency is of small importance, and the first consideration is reliability. Pumps are generally furnished in duplicate. It can be assumed that a dock is emptied 2000

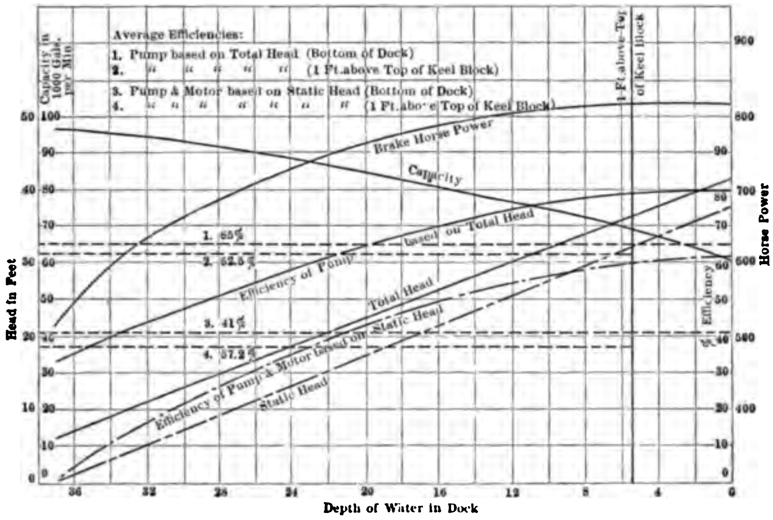


Fig. 109. Performance Curves of 54-inch Dry-dock Pump.

times in ten years. With this as a basis, each per cent of increased efficiency would show the value of the pumps of high efficiency, as demonstrated towards the end of this chapter.

The two types of docks are the graving or dry dock and the floating or pontoon dock. The former has a permanent pumping station located at a convenient place near the dock. The station is usually in a depressed chamber in the ground, so that no projections are encountered in handling the hawsers of the ships. The arrangement of piping is usually made as short as possible to avoid losses.

The floating pontoon dock is of a somewhat different character, and may be either in sections or in one unit, arranged so that, when a vessel is to be docked, the pontoon is submerged by admission of water into its compartments to pass it under the vessel. The pumps then empty the dock compartments until the vessel and dock rise sufficiently to expose the vessel so that work can be done.

The centrifugal pump has always been considered ideal for such purposes, and the low cost of installation, when compared with that of reciprocating pumps, makes its use universal.

The motive power may be either a steam engine or electric motor, depending upon the installation. Engineers usually adopt an electrically driven

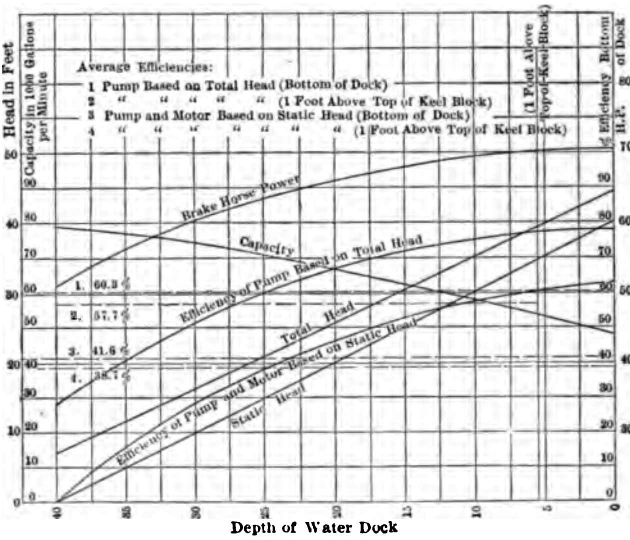


Fig. 110. Performance Curves of 45-inch Dry Dock Pump.

pump on account of the convenience with which current can be had for operating.

The time taken for emptying large docks varies from 1½ to 2½ hours. Fig.

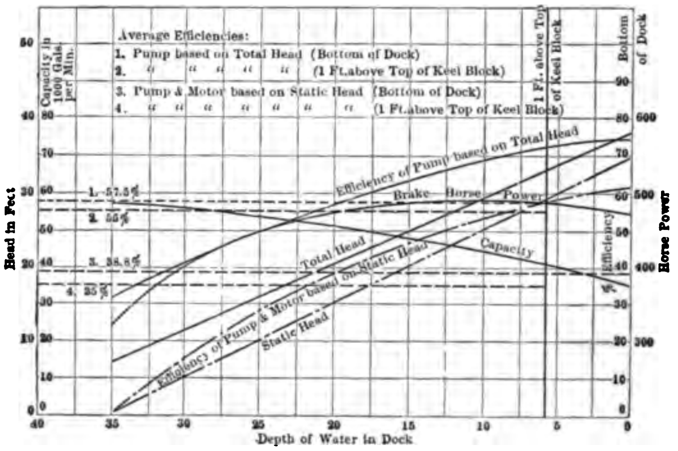


Fig. 111. Performance Curve of 36-inch Dry Dock Pump.

109 shows the curves of a 54-inch pump, Fig. 110 of a 45-inch pump, and Fig. 111 a 36-inch pump. These show the changes in pumping conditions

and the variation in capacities and heads under constant speed with the almost constant horse power necessary to prevent overload on the motors. It is evident from an examination of these figures that the problem is a

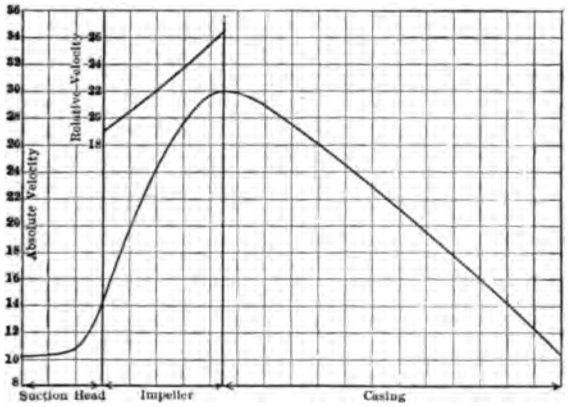


Fig. 112. Velocities in Single-entrance Impellers for Dry-dock Work.

difficult one, as the design of the impeller must take into account the variation in area of the dock at its various levels, and the outside tide-water levels, and at the same time the varying power required from no head to full

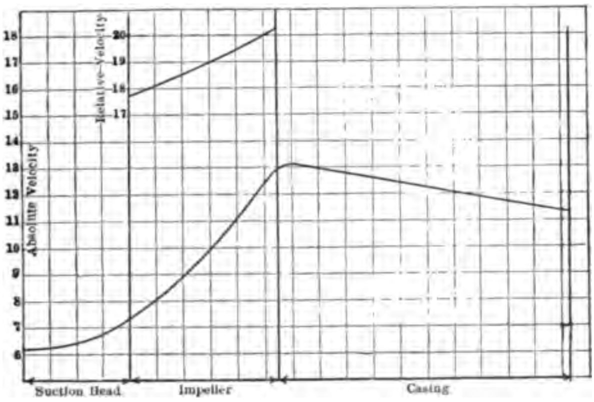


Fig. 113. Velocities in Double-entrance Impellers for Dry-dock Work.

head must not overload the motor. The curves show how it can be accomplished.

Vertical pumps having one suction entrance may not give as good efficiencies as horizontal ones with double-entrance pipes, as these allow a lower velocity for the same amount of water. Fig. 112 shows graphically

the velocities in a single-entrance vertical pump, which can be compared with the ones in a same size horizontal or vertical pump having double-entrance pipes, shown in Fig. 113, the capacities and heads being the same.

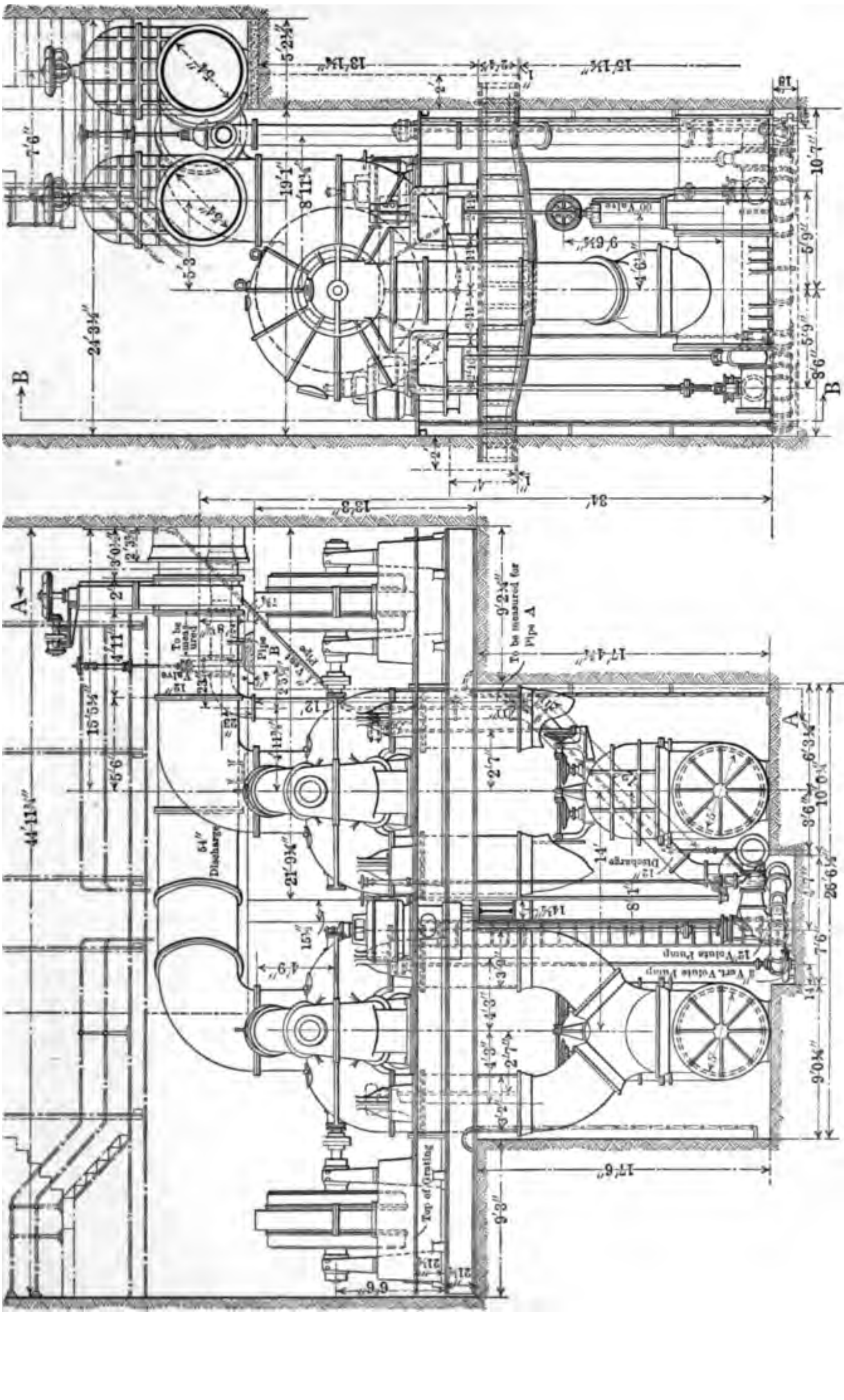
NEW DRY-DOCK AT NORFOLK NAVY YARD.

The new dry-dock and pumping equipment at Norfolk Navy Yard were completed in 1909. The pump house is located at the side of the dock and below the surface of the ground, as shown in Fig. 114. The dock is known as No. 3, and the pump well is complete with roof, galleries, stairs, mezzanine floor, and ladders to the suction pit, to give easy access to all parts.

Owing to the fact that no anchor bolts could be secured to the walls or well bottom, it was necessary to carry the entire weight of the pumping machinery on cross beams and on trusses set into pockets in the side walls, which are clearly shown in Fig. 114. In running, no vibrations are set up, which proves the ability of the structure to easily support the entire weight. The plant comprises two 54-inch double-suction volute pumps, and two 12-inch drainage pumps, including all gate valves, operating gears, and motors. The main motors are 550-horse-power three-phase 60-cycle 220-volt induction motors, operating at a maximum speed of 200 r.p.m. when the secondary windings are short-circuited. The rotors are equipped with collector rings, and full controller and resistances for starting under full-load torque, with only full-load current, and operated continuously at any speed from 75 per cent to full-load speed. The motor frames have ventilating openings allowing free circulation of air around the windings. The windings are covered with impregnated insulation to prevent damage from dampness, and have been tested to stand an alternating electromotive force of 5500 volts for one minute. The shafts are open-hearth steel and all bearings are self-oiling and self-aligning, with ample surface to insure cool running. The rise in temperature in the motors does not exceed 40° C. with the surrounding air at temperature of 25° C. Fig. 82 shows the characteristic curves of the motors.

The drainage-pump motors are of the constant-speed, 60-cycle three-phase, 220-volt induction type, with a normal output of 60 horse power at a speed of 514 revolutions, and are provided with starting devices so that the motors can be gradually brought up to full-load speed and full-load torque with no more than full-load current. These motors possess the same details of winding, ventilating devices, shafts, and bearings as the others, and were subjected to the same temperature tests.

There is also provided a transformer bank, consisting of three 50-kilowatt transformers for operating the capstan and valve motors, which are of the 2300–230-volt oil-insulated single-phase type. These were operated for twelve hours continuously with 2300 volts primary, at the rated output in amperes and a unity power factor, and heated to only 35° C. with



Section on Line B-B

Section on Line A-A

Fig. 114. Plan of No. 3 Dry-dock and Pumping Equipment at the Norfolk Navy Yard.

the surrounding air at 50° C. At the end of the run the load was taken off and the rise in potential did not exceed 40 volts. An alternating electromotive force of 10,000 volts was applied for five minutes between primary and secondary, the latter to the core, together with an alternating electromotive force of 4000 volts, momentarily applied between the low-tension winding and the core. Each transformer was also operated for two hours continuously with 2300 volts primary and an output of 50 per cent in excess of normal.

There is also provided a complete lighting transformer of the 5-kilowatt 2300-volt, oil-insulated single-phase type, for operating motor-driven fan and lights.

A full switchboard of four panels carried on suitable iron framework, two panels having three-pole single-throw 300-ampere automatic oil switches and 250-ampere ammeter for the 550-horse-power motors and similar ones for the 60-horse-power motors. The third panel controls the bank of three 50-kilowatt transformers and the fourth controls the capstan and lighting as well as the valve motors.

The gate valves for the pumps consist of two 60-inch suction valves and two 54-inch discharge valves, besides smaller valves on the drainage pumps. They are of the double-gate type, with iron bodies flanged at both ends and bronze fittings throughout. Each valve is operated by three-phase 60-cycle constant-speed 230-volt induction motors, working through cone friction clutches, fitted with an adjustment, so that, in case of obstruction in the movement of the gate valves or failure of the limit stops, the friction clutches will slip without overloading the motors. The arrangement of the clutches is such that the motors can be started light and the loads applied gradually. All of these valves are arranged to be operated also by hand through extension stems and hand wheels, and are fitted with limit stops and indicators. The drainage pumps are fitted with 16-inch valves on the suction and 12-inch valves on the discharge, with equipment similar to that of the large ones.

The main pumps are of the double-suction volute type, horizontally arranged on the shaft and directly connected to the 550-horse-power motors through special couplings. The motors and pumps are mounted on continuous bedplates for proper alignment. The pump casings are in halves parted on the horizontal line in order to give easy access to the internal working parts. Suitable manholes are provided on the casings to facilitate internal inspection and cleaning. The suction elbows are made in halves to allow the removal of the impellers and shafts without disconnecting the couplings from the shafts. The bronze shaft bearings are in halves fitted to the hubs of the suction elbows, and are provided with lubricating devices. The main casings are of the volute type, with a diffusion throat designed to give the maximum of efficiency. The casings are ribbed to

withstand any stress to which they may be subjected. The impellers are of the inclosed type, with passages connected to the two suction elbows, turned and polished all over to minimize skin friction, and are balanced so as to run true. The shafts are of the best nickel steel, ground and highly finished, and fitted with couplings at one end to connect to the motor. The bearings and stuffing boxes are composition-bushed and provided with lubricating devices.

The drainage pumps, smaller in size and of vertical form, possess the same details of construction.

The main pumps were each required to handle an average of not less than 68,000 gallons per minute when starting against a static head

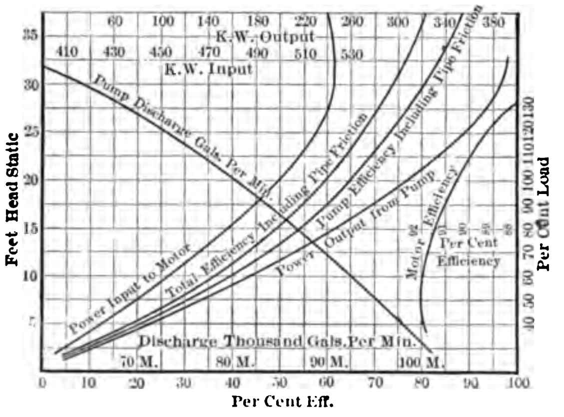


Fig. 115. Performance Curve for Total Plant Dry-dock Work.

of zero feet and ending with a static head of 36 feet through the system of piping. The power delivered by the motor to the impeller shaft at

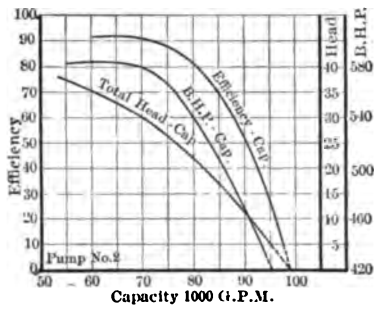


Fig. 116. Pump Performance Curves for Dry-dock Work.

no time could exceed 550 horse power. The average actual capacity of the pumps was 76,000 g.p.m. against a total head of 36 feet. The

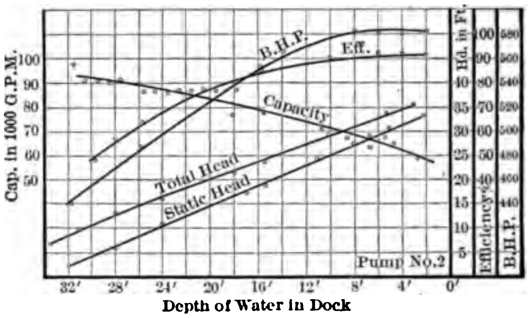


Fig. 117. Pump Performance Curves for Dry-dock Work.

dock, with a capacity of 17,000,000 gallons, was emptied in two hours. Fig. 115 gives the performance of the entire plant as tested at the Navy Yard, Norfolk, Va. The individual performance of these pumps is shown in Fig. 116, giving the characteristic curves as usually laid out; Fig. 117

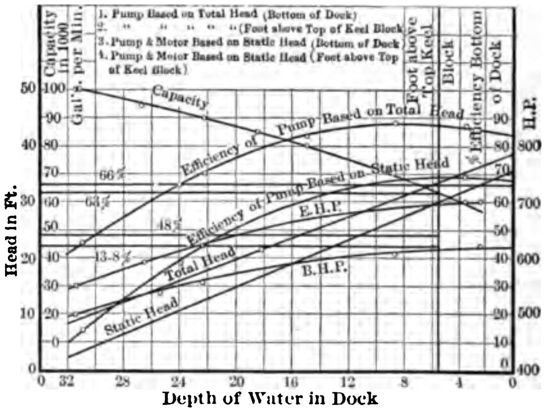


Fig. 118. Pump Performance Curves for Dry-dock Work.

shows the characteristic curves based on dock levels; Fig. 118 shows the average performance for the whole unit, including motors and piping. These curves show the exceedingly high efficiency of 92 per cent for the

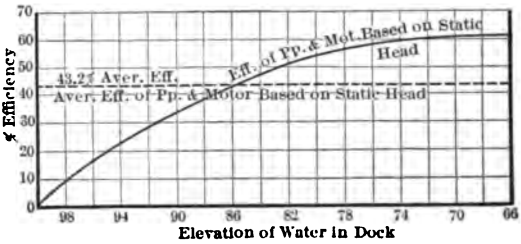


Fig. 119. Pump Performance Curves for Dry-dock Work.

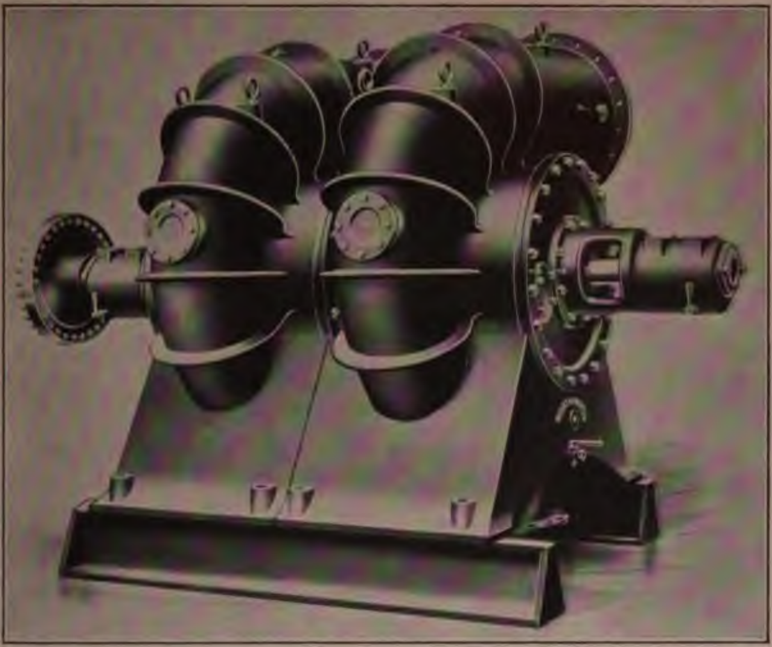


Fig. 120. Recent Type Dock Pump at Tees Dock, Middlesbrough, England.

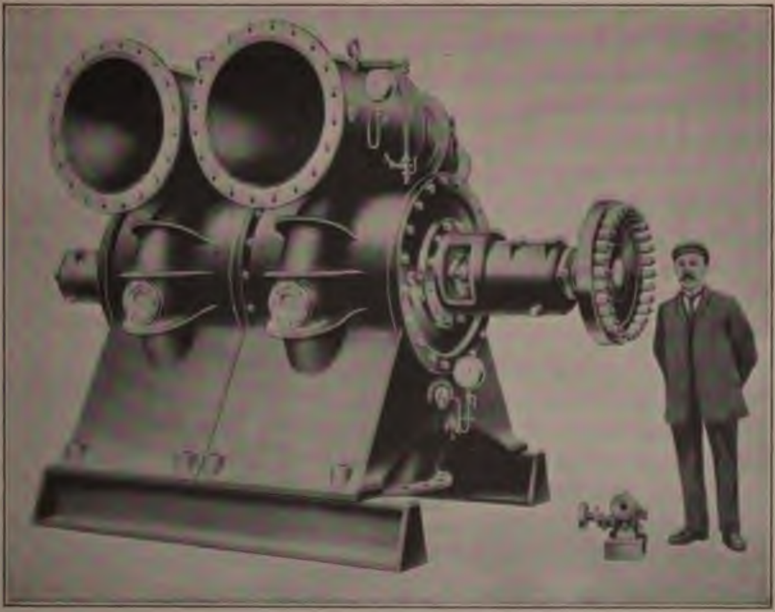


Fig. 121. Another Type of Dock Pump.

pump alone, with allowance for frictional resistance through the piping and valves. The efficiency, including pipe and valve friction, reaches 85 to

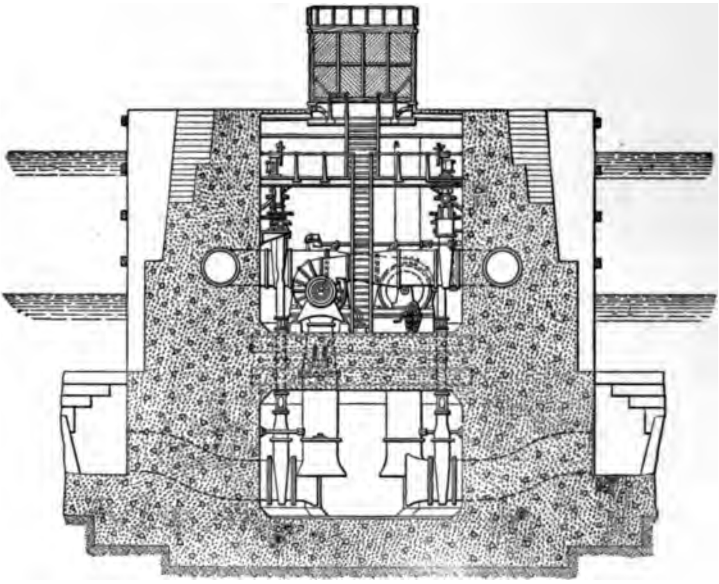


Fig. 122. Plan of Installation at Tees Dock, Middlesbrough, England.

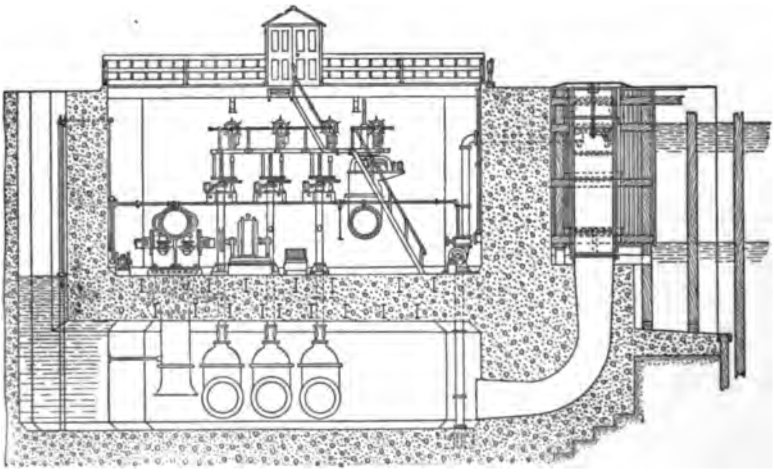


Fig. 123. Plan of Installation at Tees Dock, Middlesbrough, England,

88 per cent for a capacity of 68,000 g.p.m. at about 30-foot head, and remains practically constant up to total head of 36 feet, the capacity dropping to 62,000 gallons.

This installation is the most comprehensive and up to date in dock pumping, and forms a valuable addition to the Navy, as it provides for the docking of the new large battleships.

Another dry-dock performance from the Portsmouth Navy Yard is shown in Fig. 119, showing an average efficiency of 43.2 per cent for motors, pumps, and piping, including all losses.

Representative types of dock pumping, illustrating some later developments, are shown in Figs. 120, 121, 122, and 123, the first two showing 30-inch pumps, and the latter two an installation at Tees Dock, Middlesbrough, England, consisting of 48-inch pumps with a mean capacity of 42,000 g.p.m. and 34 feet total head at 295 r.p.m. These pumps gave a maximum efficiency as high as 85 per cent for the pumps themselves, and the power of the motor at the lowest head did not exceed by more than 16 per cent that required at the point of best efficiency. Each pump is operated by a 400-horse-power three-phase motor.

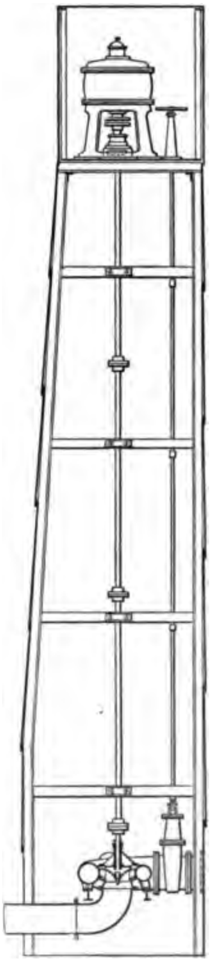


Fig. 124. Arrangement for Pontoon or Floating Dock Pump.

Fig. 124 shows the usual arrangement for pontoon or floating dock pumps, where the pumping conditions vary from those of graving docks, because the upper surface is uncovered and the weight of the ship is supported by the dock. The side sections of a pontoon dock are small and therefore quickly emptied. The difference in the level of the water within the dock and without is to be taken into account, as the head curve shows an increasing lift from the start until the floor is uncovered, then a rather quick reduction of lift, which gradually increases until the dock is emptied.

Fig. 125 shows part of the installation at the League Island or Philadelphia Navy Yard, consisting of four 54-inch units, mounted in pairs on one continuous bedplate. Fig. 126 shows a vertical installation for a Japanese Navy Yard, containing three 48-inch pumps.

Careful construction of the integral parts of such pumps makes it possible to obtain efficiencies as high as 90 per cent under the favorable conditions of dock work, handling a large amount of water, and demonstrates that this type is unquestionably the most economical one. It is in such installations that the centrifugal pump is at its best, and results can be obtained which cannot be surpassed by any other type.

Fig. 125. Part of Installation at Philadelphia Navy Yard.

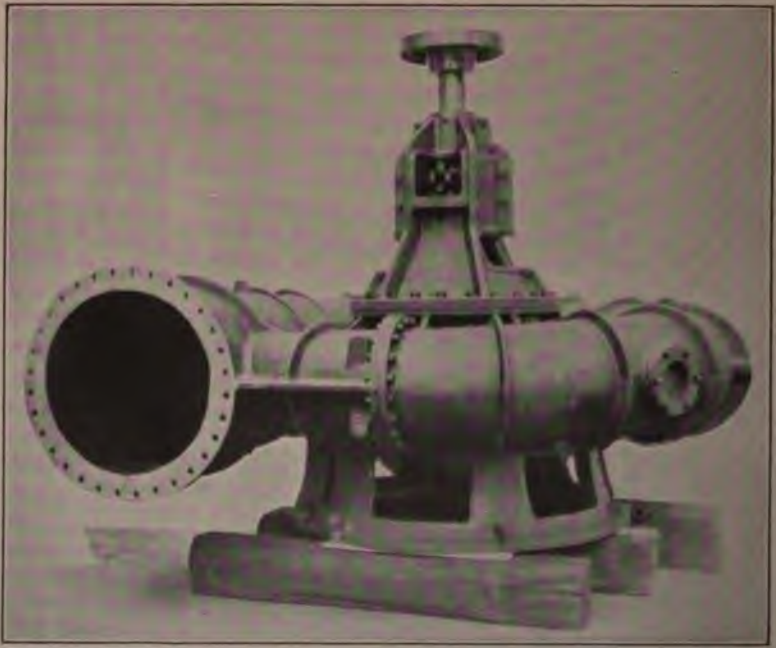


Fig. 126. Vertical Pump at Japanese Navy Yard.

CHAPTER XX.

CENTRAL FIRE-STATION SERVICE.

In fire service, centrifugal turbine pumps have been a success from the beginning. This is largely due to the fact that they have been built to suit special requirements, which have been carefully studied out by the insurance companies and their board of inspection. Much of the trouble found in the installation of a turbine pump is due to insufficient or incorrect knowledge of the exact requirements and pumping conditions. Appreciating this fact, the Associated Factory Mutual Fire Insurance Companies developed a set of specifications covering the essential features under which many pumps have been built which are giving good results.

The essential characteristics of Underwriter turbine fire pumps are ruggedness and strength, liberal water passages, noncorrosive material for all working parts, ease in dismantling, and certain special features for fire fighting. These are not secured at any sacrifice of simplicity and reliability, and with the help of such specifications the actual responsibility as to design and efficiency lies with the designer and manufacturer.

Each manufacturer must have the drawings and details approved and a sample of the pump carefully tested out at the factory under supervision of the underwriters. After this the manufacturer must agree that all subsequent pumps shall be equal to the sample tested, and that no changes in design will be made without the sanction and approval of the board.

These pumps may be run by a motor or a steam turbine; the former, however, is subject to the risk of loss due to the interruption of the electric current and is not considered as good a fire risk. In order to make this satisfactory, the source of electric power should be made as reliable as a steam supply.

There are four standard sizes of these pumps of the following capacities: 500, 750, 1000, and 1500 gallons per minute. They are made for speeds suitable for the standard motors and steam turbines. The efficiencies required are between 50 and 70 per cent, depending upon the size, and are considered reasonable.

Each pump is required to discharge a certain portion of its full capacity against a high pressure and considerably more than its capacity against a pressure of 75 pounds without overloading the motor more than 25 per cent. The higher pressures are needed for high buildings or for fires at distances which require long lines of hose. The pumps will give the neces-

sary pressures within a certain range, with constant-speed motors, but it is better to use variable-speed motors.

The pumps should have the supply under a head for priming, as they will not pick up their suction water. There is a possibility of making an automatic source of priming supply. In any case, for fire purposes a special priming tank of about four times the capacity of pump casing and pipes should be installed. On long suction pipes and large installations an independent motor-driven air or vacuum pump should be connected to the pump casing for exhausting the air. Where a reliable supply of steam or compressed air is available, an ejector or exhauster may be used, but care should be taken to have it properly proportioned for its service.

Fig. 127. Standard Underwriter Type, a 1000-gallon Fire Pump with Waterproof Motors.

When foot valves are used they should be of the multiflap type, with an area of at least 150 per cent of the pipe area, and the flaps should open toward the sides of the valve so as to give an unobstructed opening. Screens should have a clear area of 200 per cent of the pipe area.

If check valves are used on the discharge, they should also have the clapper or valve folding back against the wall.

The motors should not burn out if all the streams are shut off or when they are opened up to more than the capacity of the pump, and they should easily stand a load of 25 per cent over that for which the pump is built, and should be protected from possible leakages from the pump.

The control affected by throttling, speed variation, or by both, should be

considered. The throttling of a pump should be done in the discharge valve and not in the suction valve. Speed control is more satisfactory and can be obtained with both motors and steam turbines. There are several classes of electric motors used for fire pumps, — alternating-current motors of constant-speed induction type, and direct-current motors. In the induction type of motor, with the pump discharge valve closed, the torque required is about 50 per cent of that for full load. From this it can be seen that the motors may be readily started on about 65 per cent of the regular voltage and brought up to speed with not more than double the full-load current. Full voltage can be obtained with a little increase in current and the valve opened for the desired delivery.

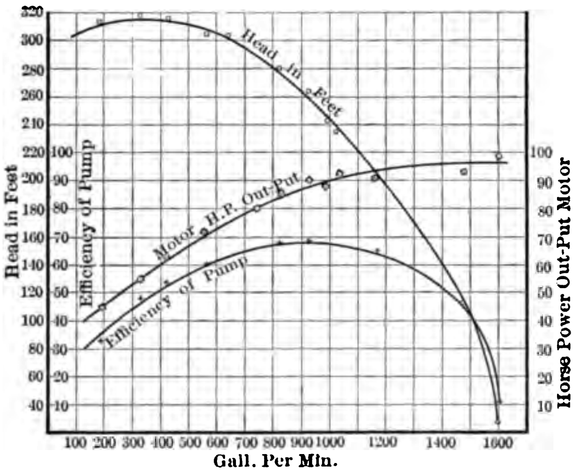


Fig. 128. Performance Curve Underwriter Pump.

The method of regulating shunt-wound or direct-current motors of variable speed is simple and well understood, and the variable-speed induction motor is similar to the direct-current armature in control.

Fig. 127 illustrates a 1000-gallon fire pump with waterproof motors of the standard Underwriter type.

Fig. 128 shows the characteristics for a 1000-gallon four-stream pump, designed to meet all the requirements, and shows that on the Underwriter basis of 250 gallons per stream, $2\frac{1}{8}$-inch nozzle, the pumps would give

4 streams at 240 feet head, or 103 pounds pressure;
5 streams at 185 feet head, or 80 pounds pressure;
6 streams at 105 feet head, or 45 pounds pressure;
3 streams at 290 feet head, or 126 pounds pressure;
2 streams at 310 feet head, or 135 pounds pressure.

A maximum head of 318 feet, or 137 pounds pressure, is reached with a capacity of from 300 to 500 gallons or one to two streams. With this range it can be seen that the centrifugal fire pump can compete with the reciprocating fire pump.

The horse-power output of the motor never reaches 100, while the Underwriter requirements for this size of pump allows for 100 to 107 horse power. The theoretical horse power required to discharge one stream of 250 gallons under a pressure of 100 pounds is 14.61, to which must be added the lift of the suction supply and the losses in the pipes, or about 1.4 horse power, making a total of 16 theoretical horse power for each fire stream.

The following is the record of a test of a 1000-gallon centrifugal fire pump made under the supervision of the inspectors of the Associated Factory Mutual Fire Insurance Company, Nov. 29, 1910. The pump was designed for a 100-horse-power induction motor, but the test was made with a direct-current shop motor, temporarily connected to the pump. The speed of the motor, with no load, was given on the plate as 1200 r.p.m. Assuming some falling off from this speed under load, the test was made at as nearly 1150 r.p.m. as was practicable. The suction was taken from a submerged tank through a strainer and foot valve; the discharge was conducted to a large box with properly fitted baffle plates and a 36-inch weir. Two tests were made with three 1½-inch and four 1⅛-inch Underwriter play pipes, discharging through 50 feet of cotton rubber-lined hose into the weir box, and they checked up reasonably close with the weir measurements. The revolutions were obtained by an English tachometer checked up occasionally by an ordinary speed indicator. The pump was run without stopping for about an hour and a half. No difficulty was experienced with any of the bearings. The behavior of the pump under all loads, from no load to overload, was very satisfactory. The use of three stages in the construction of this pump makes it possible to secure a much higher pressure when running at half capacity than would be possible were only two stages employed.

The pump was tested up to 240 pounds pressure and showed no weakness. Subsequently the pump was opened and the first impeller and section of casing removed.

The results of the test are given in the accompanying table. The capacity of 1000 gallons per minute was obtained under a total head of 236 feet at 1150 r.p.m. and with an efficiency of 66½ per cent. The motor horse-power output drops off at the higher discharges, so that it is not likely that the motor could be overloaded. The electrical readings for computing the efficiency of pump were taken from Weston instruments used in the regular testing work of the shop.

One of the first large cities to install this type of fire pump was **Brooklyn, N. Y.**, followed by **New York and San Francisco. Philadelphia, Winnipeg,**

TABLE NO. 4.—TEST OF 1000-GALLON CENTRIFUGAL FIRE PUMP.

Made by HENRY R. WORTHINGTON, *Nov.* 29, 1910.

Number.	Revolutions per minute.	Pounds gauge pressure at pump.	Feet suction to pump gauge.	Total head in feet.	Weir head in feet.	Voltmeter.	Ammeter.	Horse-power input to motor.	Efficiency of motor.	Horse-power output from motor.	Horse-power output from pump.	Efficiency of pump.
1	1155	130	4.6	304.6	....	230	145	44.8	73.5	33	....	....
2	1150	131	7.8	310.8	.190	230	188	58	78	45.2	14.9	33
3	1150	134	7.8	317.8	.334	240	215	69.1	80	55.2	26.7	48.4
4	1150	133	8.1	315	.425	240	240	72.2	81	62.5	33.8	54
5	1150	128	8.8	304	.567	241	270	87.2	82.4	72	43.4	60.4
6	1160	127	9.1	302	.644	240	285	91.7	82.5	75.6	50	66.1
7	1160	117	10.1	280	.824	240	320	103	83	85.5	58.1	68
8	1155	110	11.1	265	.925	240	335	108	83	89.5	61.7	69
9	1150	100	11.4	242.4	.992	239	340	106.5	83.5	89	60.7	68
10	1150	97	12	236	1.032	240	345	111	83.5	92.6	61.5	66.5
11	1145	82	13.1	203.1	1.170	230	355	109	83.5	91.2	59.8	65.6
12	1150	41	17.1	112.1	1.480	230	365	112.5	83.5	94	41.8	44.5
13	1150	4	18.1	27.3	1.600	240	370	119	83.5	99.5	11	11

and the New York branch at Coney Island installed gas-engine-driven triplex reciprocating power pumps. A brief description of the first high-pressure fire-service turbine pumping plant installed at Brooklyn, N. Y., will be of interest.

This consists of two separate stations, one situated at corner of Willoughby Avenue and St. Edwards Street, and comprises three units. The other or main station is at the northeast corner of Furman and Joralemon Streets and has eight units. Unlike the New York high-pressure fire-service stations, both these plants can secure water from either a salt-water or a fresh-water supply, requiring a more careful design in order to give the same results when working with a suction head of 50 pounds pressure and when lifting the suction water from a supply 20 feet below the pumps. The salt-water supply comes from the East River, and the fresh-water from a 20-inch main. This arrangement is of great importance, as it insures a fire stream even though the city main that supplies the suction water should break. It is usually required that a pumping station relying upon a salt-water supply should be fitted up so that equally good results can be obtained with salt water or fresh.

In installing the high-pressure fire-service stations in the Borough of Brooklyn, equal efficiency was demanded under both conditions.

The main station at the foot of Joralemon Street consists of eight units, as shown in Fig. 129. The relief station at Willoughby Street is shown in Fig. 130. The pumping units are all identical, and interchangeable. Fig. 131 shows one of the units complete. It consists of a six-stage turbine pump directly connected to the motor. The pumps are of the horizontal type, each capable of delivering 3000 gallons per minute, when operating at 735 revolutions and delivering against a pressure of 300 pounds per square inch, with a suction lift of 20 feet, or under a supply pressure of 50 pounds per square inch or less when taking supply from the 20-inch main. The size of the suction openings for each pump is 12 inches and the discharge 10 inches. These pumps so operate that the pressure system can be regulated between 100 to 300 pounds, by increments of 50 pounds, the speed remaining constant, the regulation being effected by the design of impeller and by special regulating valves. The brake horse power of the motor when operating at any lower pressure than 300 pounds does not exceed that at 300 pounds, thereby obviating overloading the motor when the capacity increases and the pressure drops.

Each pump is directly connected to a three-phase 25-cycle 6000-6300-volt 800-b.h.p. motor of the induction type, and fitted with all the necessary starting and controlling devices. The full-load efficiency of these motors is 95 per cent, the power factor 96, and the slip 2 per cent. At three-quarters load, the motor efficiency is 95 per cent and the power factor 95½ per cent.

Fig. 129. Main Pumping Station at Foot of Joralemon Street, Brooklyn, N. Y.

Fig. 130. Relief Pumping Station at Willoughby Street, Brooklyn, N. Y.

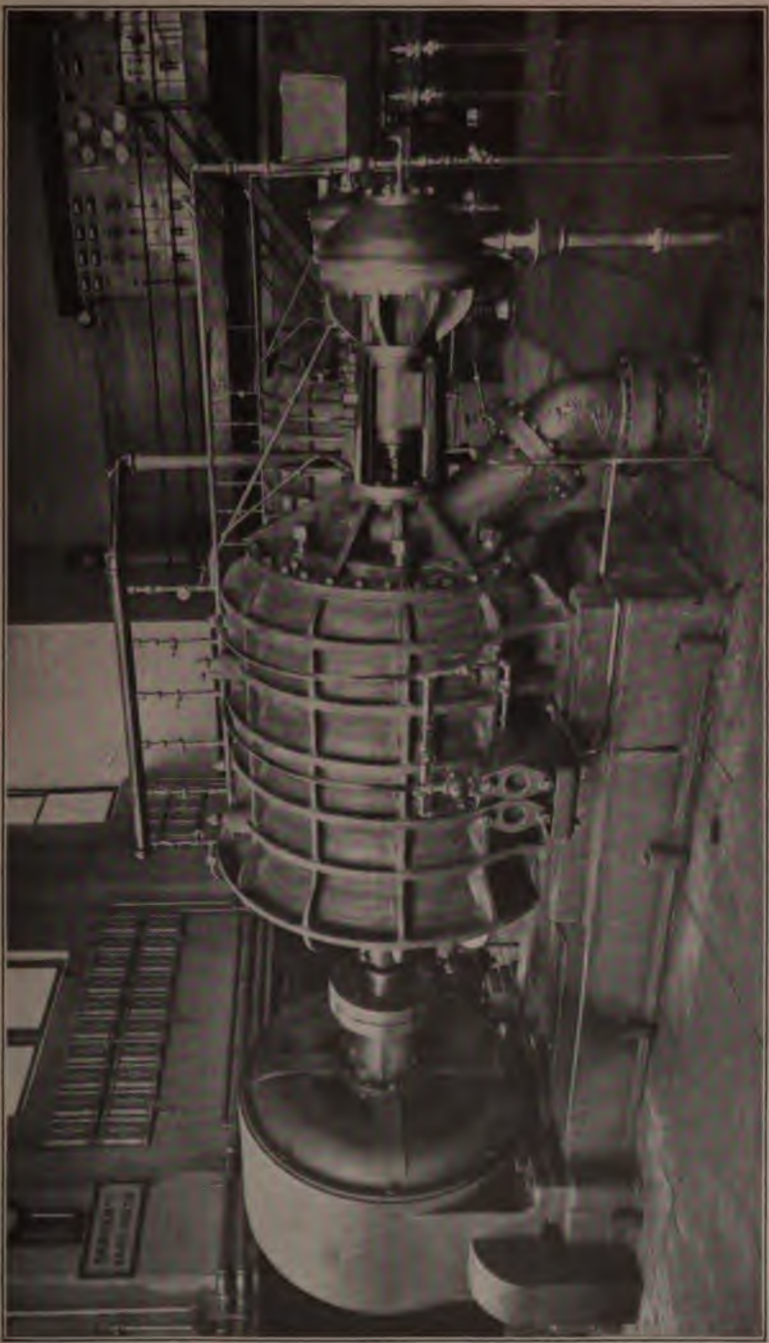


Fig. 131. One of the "Worthington Units" Used in the Main and Relief Pumping Stations in Brooklyn, N. Y.

The results of the tests given in Figs. 132, 133, and 134 show an efficiency on salt water of 75 per cent and on fresh water of 76 per cent for the pumps,

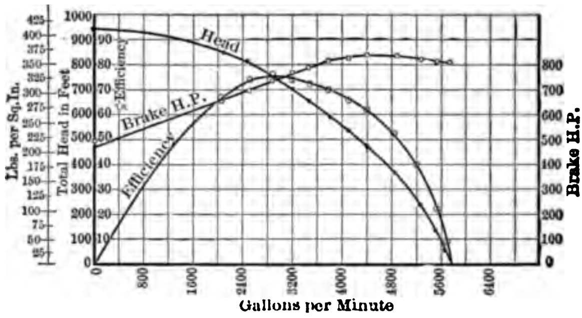


Fig. 132. Performance Curve for Brooklyn Fire Pump.

and an over-all efficiency for motor and pump of 72.2 per cent. The water was measured by a Venturi meter in the mains. The capacity of the pumps at various pressures is shown in the curves, as well as the horse power required for operation, and the characteristics controlling them.

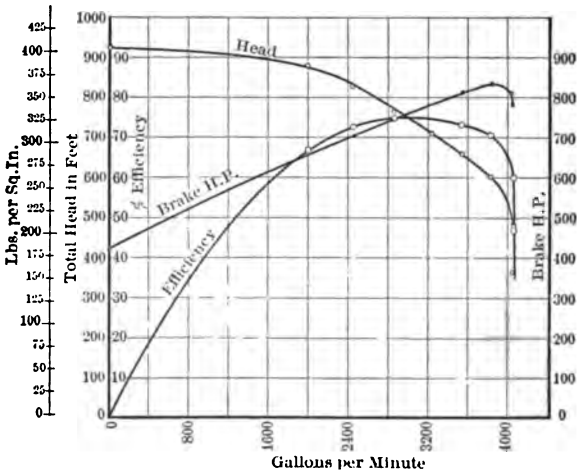


Fig. 133. Performance Curve for Brooklyn Fire Pump.

When operating with salt water the pumps are primed by a motor-driven vacuum pump operating automatically. It was found that with either salt or fresh water the pumps can be started and brought to full speed in less than 45 seconds, without having the starting current exceed 150 per cent of the full-load current of the motors. An additional time of 10 seconds was required to bring the pump up to a pressure of 100 pounds. Therefore, a unit can be started from complete rest and brought up to

speed under a water pressure of 100 pounds in less than one minute. The pressure can be varied from 300 pounds down without overloading the motor, an extremely severe condition.

It is interesting to note the results obtained by gas-engine-driven power pumps for similar purposes, notably those at Philadelphia and Winnipeg. The mechanical efficiency at Winnipeg is from 77 per cent to 82 per cent on pumps of 1800 gallons per minute capacity, and the pressure 300 pounds, giving a B.t.u. efficiency per indicated horse power of 8600, and on water horse power of 11,774. The Philadelphia plant gave 11,160 B.t.u. per indicated horse power and the Coney Island plant gave 12,682 B.t.u. per water horse power. Such gas-engine-driven fire pumping equipments operated on producer gas will give from 180 millions to 210 millions duty in foot

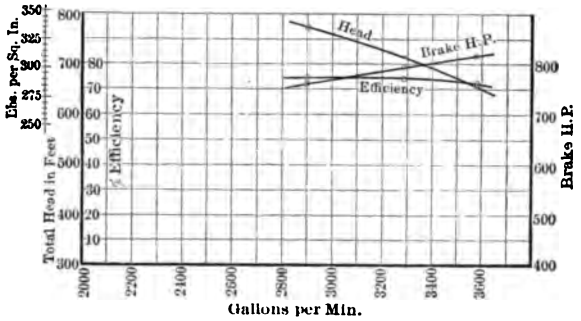


Fig. 134. Performance Curve for Brooklyn Fire Pump.

pounds of work done per one million B.t.u. consumed, which is a very low cost for operating, but as they are not operated continuously this is not of great consideration. They have the advantage of low cost of installation and operation, but these are offset by the flexibility of turbine-pump installations in handling larger capacities at any desired pressure.

Electrical or turbine-driven centrifugal-pump fire stations are considered more reliable than gas installations, and the question of economy in operation is secondary, because the total cost of pumping is only a small portion of the running expenses.

The capacities of the various stations built are as follows: Brooklyn, 24,000 gallons per minute at 300 pounds pressure and 40,000 gallons at 150 pounds pressure; New York, 30,000 gallons; Coney Island, 4500 gallons; Philadelphia, 9000 gallons; San Francisco, 20,000 gallons; Winnipeg, 9100 gallons. From this it will be seen that the gas pumping engines are of considerably smaller capacities. The requirements in all such stations should be for a definite capacity and head, for although it might seem that an excess capacity is good, it usually overloads the motors, and extra power has to be paid for.

High-pressure fire-service stations are becoming a necessity in this country owing to the high buildings and the larger volumes of water required in fighting fires. A pressure of 300 pounds is usually needed to overcome frictional losses and supply $2\frac{1}{2}$- to 3-inch nozzles at a pressure of 100 pounds. Portable fire engines to handle such conditions are entirely out of the question, and the only practical means for meeting the situation is the permanent high-pressure station.

Fire Engines. — The application of turbine pumps to automobile fire engines is an interesting development. These pumps are made in stages of one, two or three, suitable for the speed of the automobile engine, and the capacity varies from 250 gallons to 1000 gallons for pressures of 220 pounds down to 120, when delivering water through 1000 feet or 500 feet of $2\frac{1}{2}$-inch hose and $1\frac{1}{8}$-inch nozzles. A usual requirement is that a pump should deliver 700 gallons per minute against a pressure of 120 pounds and 300 gallons against a pressure of 200 pounds.

The importance of obtaining the highest pressure is apparent on account of internal friction in the hose which reduces the effective pressure at the nozzle. The regularity of flow and absence of water hammer make this the best solution of a fire pump for a portable fire-fighting apparatus.

The action of a turbine pump allows the shutting off of any particular hose, nozzle or all, without tendency to burst the hose as may be the case with reciprocating or rotary fire pumps. The pump is considerably lighter and takes less space, requiring no suction or discharge air chambers. The pumps are designed to fit on the back part of the chassis of an automobile to be driven by a shaft directly from the engine or through transmitting gears.

The starting up when a heavy suction lift is encountered is accomplished by means of a small rotary vacuum pump operated by a magneto on the shaft of the engine, which exhausts the air in pump and pipes. When the turbine pump is primed, it is thrown into action by the clutch and develops immediately its head pressure and the small vacuum pump is then cut out. A special by-pass and valve are attached to the pump to allow the water to pass from discharge to suction in a small stream so as to prevent heating should all nozzles be shut off and the clutch left in while the engine is running.

It is therefore apparent that rapid progress indeed has been made in adapting the turbine pump to an increased field, giving a better and more efficient fire engine. In large cities the automobile turbine fire engine is usually designed to carry nothing but the pump. In smaller towns a combination of fire engine, chemical and hose cart is desirable, as the most useful apparatus for a fire department at minimum upkeep cost. The economic advantages of such a combination seem beyond dispute and the success is already assured. The reliability is only a question of design

with proper power of gasoline motor, which should be between 100 and 125 horse power for the capacities stated, and well designed to stand up under continuous service without overheating. The tendency in the large cities is to supplant the fire engine in centers of business, by high-pressure systems of independent mains and hydrants, such as have been described for New York and Brooklyn, but for the suburbs and smaller cities and towns these independent fire engines will be needed.

The Underwriters have set a standard for such fire apparatus that can probably only be reached by the application of turbine pumps to the fire engines, just as the turbine has supplanted the reciprocating pump in stations for high-pressure fire mains and on fireboats.

Such automobile fire engines will weigh complete from 9000 to 13,000 pounds, and will travel at the rate of 30 to 35 miles an hour over city streets.

CHAPTER XXI.

FIREBOATS AND SHIPBOARD SERVICE.

FIREBOATS.

Turbine pumps are extensively used for fireboat equipments where reciprocating flywheel pumps were at one time considered indispensable. The first boats so equipped were the "James Duane" and "Thomas Willett," belonging to the New York fire department; since then, however, Chicago, San Francisco, Seattle, and Duluth have followed the example set by New York.

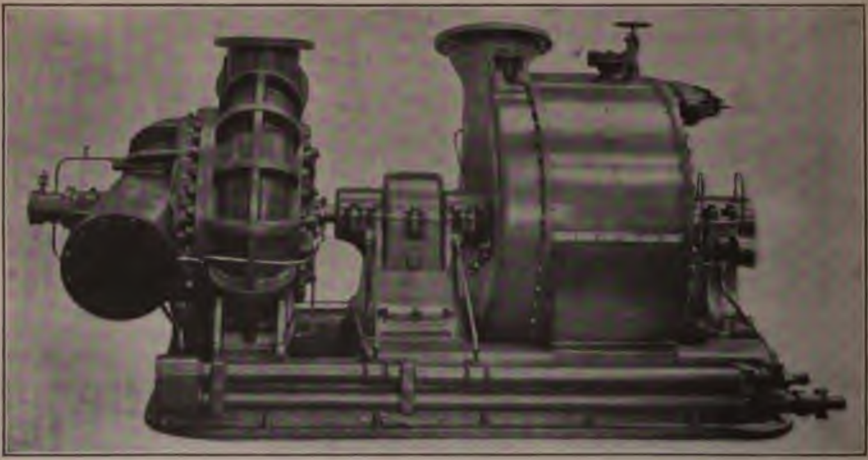


Fig. 135. Worthington Steam-turbine Pump used by New York Fire Department.

The New York system of protecting its water front adds considerably to the shipping value of the harbor, and to the value of property along the water front. Fig. 135 illustrates one of the pumps installed on these boats. The boats are duplicates in all respects and are 123 feet long, 27 feet wide, and 14 feet deep. Each boat contains two units of two-stage pumps, piped to run in series or parallel, and each boat is capable of handling 9000 gallons per minute against 150 pounds pressure, or half that capacity at 300 pounds pressure for use in fighting land fires along the water front where the distance is too great to be reached by direct streams. The pumps are driven

by 600-horse-power General Electric steam turbines operating at a speed of 1800 revolutions.

The following record of the actual tests of these boats gives the details of performance.

On the "Thomas Willett," with both pumps running at 1800 revolutions per minute, the following readings were obtained:

TABLE NO. 5.—"THOMAS WILLETT."

Pressure at nozzles.					Pressure at pumps.		G.p.m.	R.p.m.	
Size, 2 in.	Size, 2½ in.	Size, 3 in.	Size, 3 in.	Size, 3 in.	Port.	Starboard.		Port.	Starboard.
0	0	100	102	100	160	160	9,035	1800	1800
0	0	115	115	110	170	170	9,600	1800	1800
0	0	115	115	112	175	175	9,625	1900	1900
90	90	90	90	85	140	140	11,460	1900	1900
......	......	230	......	......	155	310	4,500	1780	1790

On the "James Duane" the following tests were made:

TABLE NO. 6.—"JAMES DAUNE"

Pressure at nozzles.			Pressure at pumps.		G.p.m.	R.p.m.	
Size, 3 in.	Size, 3 in.	Size, 3 in.	Port.	Starboard.		Port.	Starboard.
73	73	....	150	....	5120	1900	....
75	60	0	....	150	4920	....	1900
40	55	53	....	125	6280	....	1800
85	85	85	165	165	8280	1800	1800
90	90	87	175	175	8480	1900	1900
170	....	....	165	330	3900	1800	1800

On both boats the vacuum varied from 26 to 27 inches, and the steam pressure from 175 to 190 pounds.

The pumps ran with no vibration, and standing on the deck one would scarcely know the pumps were running except for the water thrown.

It was found that the speed for throwing full capacity could be obtained in from 25 to 40 seconds. The last test on the "Thomas Willett" consisted in putting the two pumps in series for 300 pounds pressure.

Fig. 136 illustrates the internal arrangement of these pumps and the design of impellers for high speeds and high pressures. Fig. 137 shows another size of similar type. These illustrations show turbine pumps especially designed for steam-turbine drives on fireboats, where the primary requirement is instant operation. The use of turbine pumps eliminates vibration in the boat when running full speed, and the ability to come up

to speed in 20 to 30 seconds is something never before accomplished in fire pumps. The possibility of closing down all the hose streams suddenly

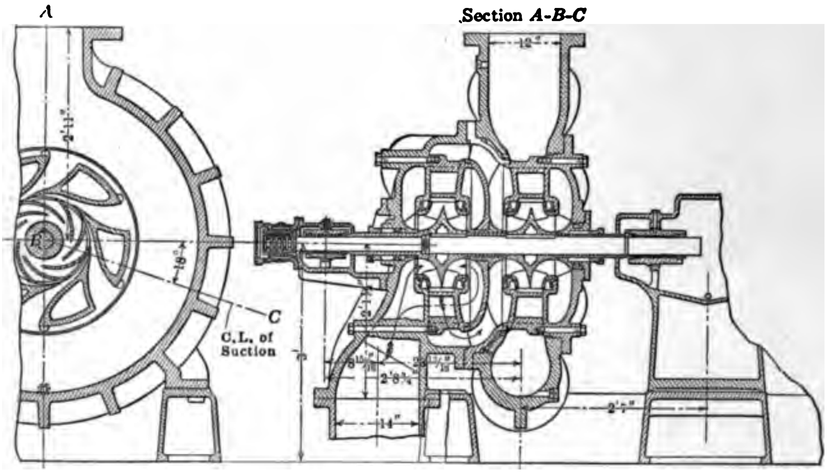


Fig. 136. Internal Arrangement of Turbine Pump used on Fireboats.

without harm is welcome to the operator, as in the reciprocating type of pumps care must be taken to slow down gradually or something will break.

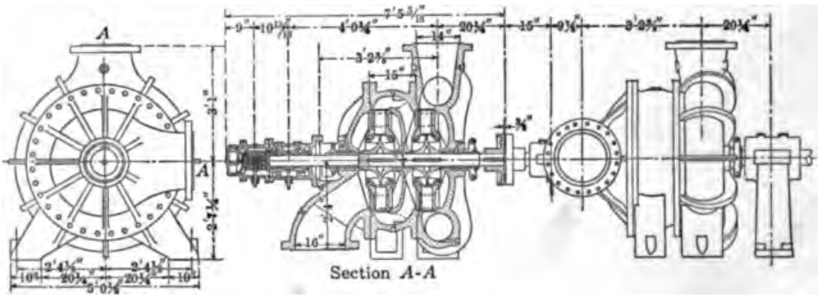


Fig. 137. Small Size Turbine Pump used on Fireboats.

The boats have demonstrated their readiness for service and have rendered valuable services in several large water-front fires. There can be no doubt that for fire service the steam-turbine-driven turbine pump is far superior to any flywheel reciprocating pump, and that in time it will replace all others.

ON SHIPBOARD.

Because of its simplicity and compactness the centrifugal pump has come into very general use on shipboard. Some of these applications have been given in the previous chapter and under "Circulating Water," and so need

no further mention here. This type is also used very extensively for handling ballast water from the double bottoms, ballast tanks, and water-tight compartments, and also in bridge work and the sanitary service. Wrecking boats are always equipped with two centrifugal pumps with about 12- or 14-inch discharge permanently fixed on board. They are engine-driven and of either the vertical or horizontal type. The boilers are usually designed to work with salt water and to stand a lot of abuse.

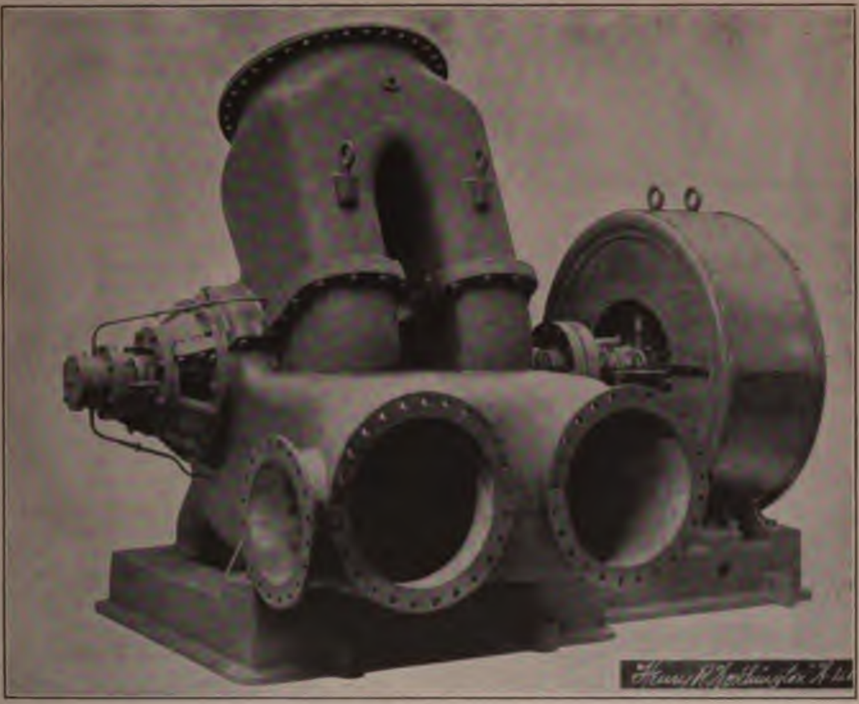


Fig. 138. One of the Latest Circulating Pumps of the Birotor Type used on the Battleship "Arkansas" of the Dreadnought Class.

Fig. 138 is a photographic view of one of the latest circulating pumps of the birotor type used in marine practice and now installed on the battleship "Arkansas" of the dreadnought class. A steam-turbine-driven circulating unit can be arranged to operate under almost all conditions, such as high pressure exhausting to atmospheric or to the condenser, or with low-pressure steam or as a mixed-flow turbine, and will furnish circulating water with absolute reliability.

It is to be noted that the impellers work in parallel, having one main inlet pipe and one outlet pipe, with a special opening on the side of the suction chamber for pumping out the bilges. The pump casing is split horizontally

in order to facilitate repairs aboard ship, where space is of paramount importance.

The compactness of the design lends itself readily to marine service, and in this instance the total length is only 11 feet for a unit having a capacity of 27,000 gallons with a maximum head of about 37 feet, including all losses through tubes, sea-cock piping, and valves. The particular feature in such a pump is the high water speed, which makes it possible to design a compact pump and at the same time to obtain the reasonably good efficiency of 66 per cent.

"The minimum requirements are 15,000 gallons per minute against a head of 25 feet, of which 15 feet are suction lift, when working on the bilges. For this service there is provided a side suction opening. The main suction openings, through which the circulating water for the condensers is supplied, consists of two separate connections with pipes to the sea cocks."

CHAPTER XXII.

SPECIAL HIGH-SPEED INSTALLATIONS.

The high rotative speed of the steam turbine, and the convenience of operating it direct-connected to a centrifugal pump, have presented an interesting problem to the pump designer. Where the quantity of water is not large a small impeller may be safely used if constructed of a material

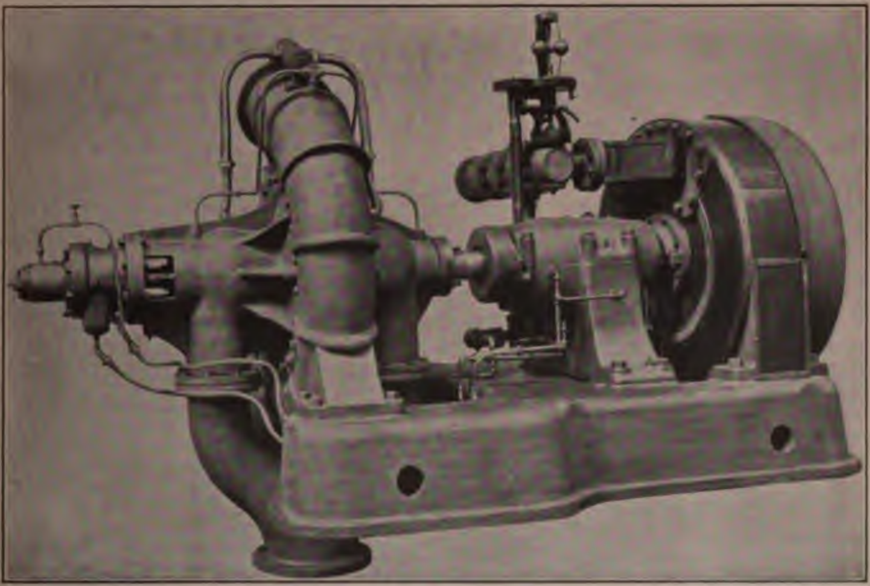


Fig. 139. Novel Design of Turbine-driven Pump.

which will not be cut by the water. Fig. 139 illustrates a rather novel design of turbine-driven pump with a normal capacity of 2300 g.p.m. to 3300 g.p.m. and a minimum capacity of 900 g.p.m. for elevator service against a head of 150 pounds pressure. The shaft is carried in two spherical bearings, one near the steam turbine and the other on the outer end of the pump, to allow the shaft to take a true position. There is no thrust in either the turbine or the pump, but for safety a special bearing has been arranged at one end. The turbine and pump are secured to a common base plate, and the unit is entirely contained.

Pumps of this design may be made with single impellers up to 300 feet

head, without the need of diffusion vanes. Efficiencies of these pumps vary from 75 to 80 per cent.

Fig. 140 illustrates a 14-inch two-stage high-speed steam-turbine-driven pump, arranged to run in series or in parallel. It was designed for 11,000 gallons per minute at 250 feet head, or 5500 gallons at 500 feet head, and is driven by a 1000-horse-power turbine running at 1600 r.p.m. The test of this unit is given in the following table:

TABLE NO. 7.—TEST OF 1000-H.P. TURBINE DIRECT-CONNECTED TO TWO 14-INCH WORTHINGTON TURBINE PUMPS.

QUEMAHONING DAM, *June* 7, 1910.

	Series.		Parallel.			
Number of test.	1	2	3	4	5	6
Steam Turbine.						
R.p.m. turbine	1,631	1,631	1,617	1,603	1,620	1,620
Steam pressure abs.	121.6	122.3	112.5	113.6	124.0	126
Quality steam, per cent.	96.5	96.4	97.1	96.4	97.4	97.5
Inches vacuum	25	25	25	25	25	25
Barometer, inches	28.5	28.5	28.5	28.485	28.485	28.5
Vacuum ref. to 29.92	26.98	26.98	27.04	26.72	27.24	27.67
Temperature exhaust	144	144	144	116.6	110	111
Hot well, deg. F.	58	58	57	58	59	60
Air-pump suct., deg. F.	50	60.6	61.5	67	68	71
Cooling H_2O ent., deg. F.	56	56	56	57.6	57.5	58.5
Cooling H_2O, leaving, deg. F.	66.5	69	64	58–64.6	60–76.8	67–85
R.p.m. air pump	80.3	81.2	80.75	85.3	80.25	83.5
Pump.						
Lift in feet	15	14.65	15.6	16.15	13.2	13.0
First stage, lbs. press.	104.8	123	55.5	100.3	135.5	136.3
First stage, ft. press.	242.0	284	128	232.0	312.0	314
Second stage, lbs. press.	219.8	257	149.2	101.0	137.2	138.0
Second stage, ft. press.	507.0	294	342.0	233.3	317.0	319.0
Dist. between gauge, ft.	3.0	3.0	3.0	3.0	3.0	3.0
Total head, ft.	525	611.5	360	251.8	330.7	332.5
R.p.m. pump	1,631	1,631	1,617	1,603	1,620	1,620
Weir reading, ft.	.594	.522	.652	.947	.388	.280
Capacity, gal. per min.	5,430	4,480	6,180	11,020	2,880	1,790
Water horse power	722	691.5	563	654	241	151
Condensed steam	20,320	16,256	12,192	12,192	12,192	4064
Dry condensed steam	19,600	15,670	11,840	11,720	11,750	3960
Dry steam per hour	20,400	19,620	16,430	19,450	13,240	8150
Dry steam per hour, water horse power	28.3	28.4	29.3	29.8	55	54

Francis formulæ used for weir calculations.

Where large quantities of water are needed, an ordinary style of pump cannot be used, since the large diameter of the impeller gives a prohibitive

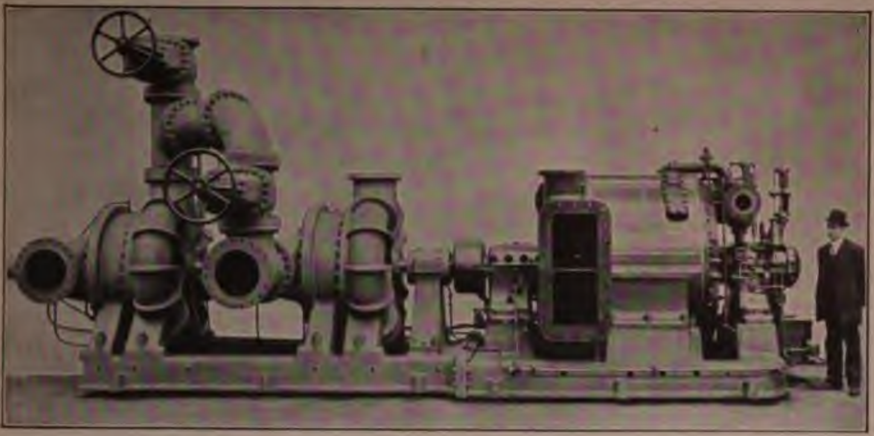


Fig. 140. 14-inch Two-stage High-speed Steam-turbine-driven Pump.

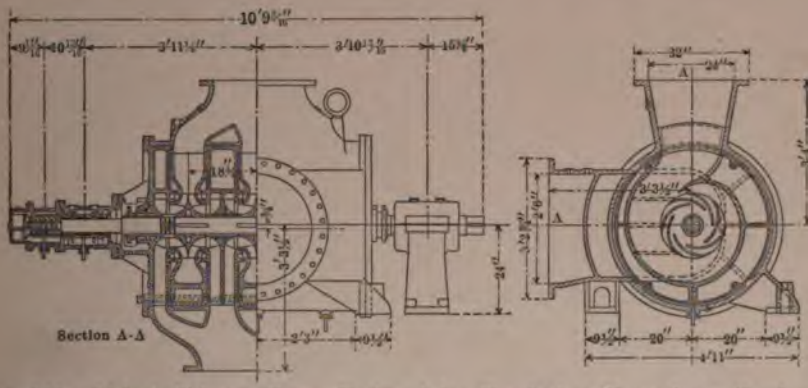


Fig. 141. 24-inch Trirotor Pump Designed for Circulating Water for a Condenser.

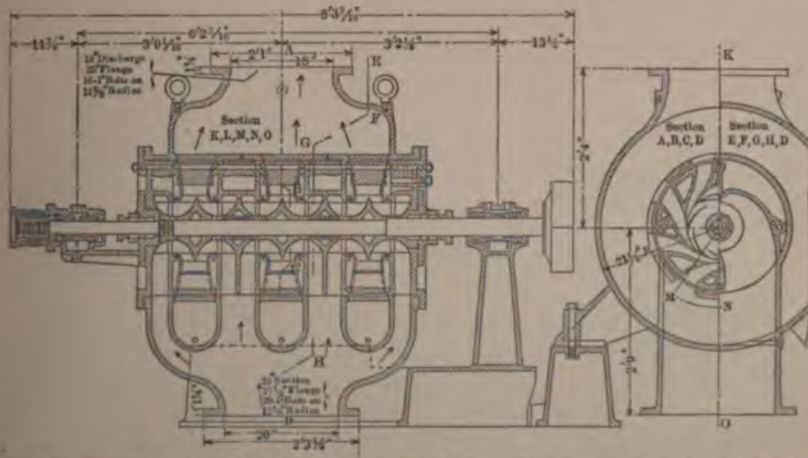


Fig. 142. Pump Designed for Circulating Water for a Condenser, Fitted with Diffusion Vanes.

peripheral velocity. For this service a multirotor pump has been designed which bears the same relation to the capacity of the pump at high speed that the multistage design does to the head.

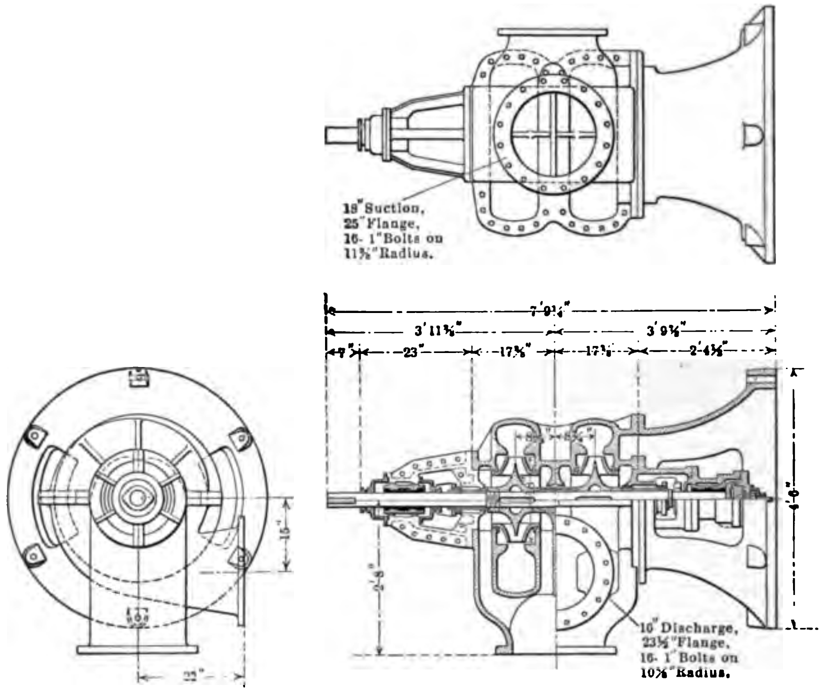


Fig. 143. Vertical Type of Birotor.

Fig. 141 illustrates a 24-inch trirotor pump designed for circulating water for a condenser. It is practically three small impellers working in parallel so that the proper linear velocities may be obtained with the high

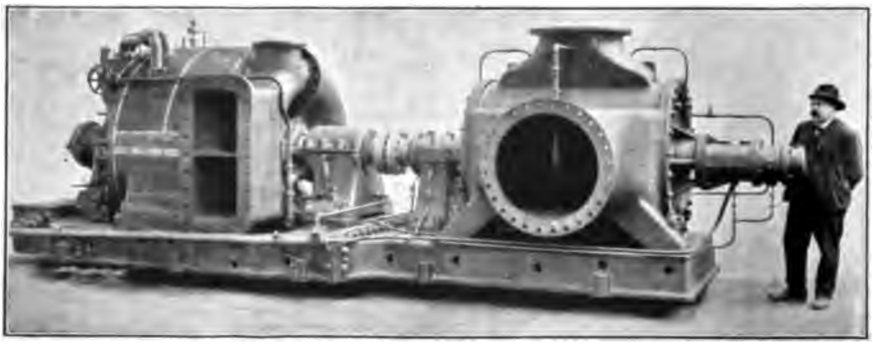


Fig. 144. A 24-inch Trirotor High-speed Unit.

rotative speed of the steam turbine. Fig. 142 illustrates a similar pump fitted with diffusion vanes in the discharge chamber and suitable for high

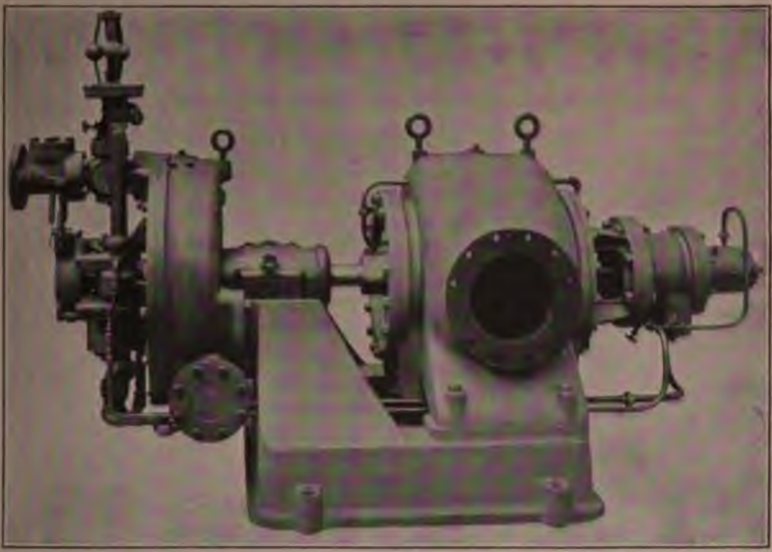


Fig. 145. A Birotor Type of Steam-turbine-driven Pump for Circulating Water in Power Stations.

circulating heads. Fig. 143 shows a vertical type of birotor suitable for certain classes of plants requiring this type.

A 24-inch trirotor high-speed unit is shown in Fig. 144. It is designed

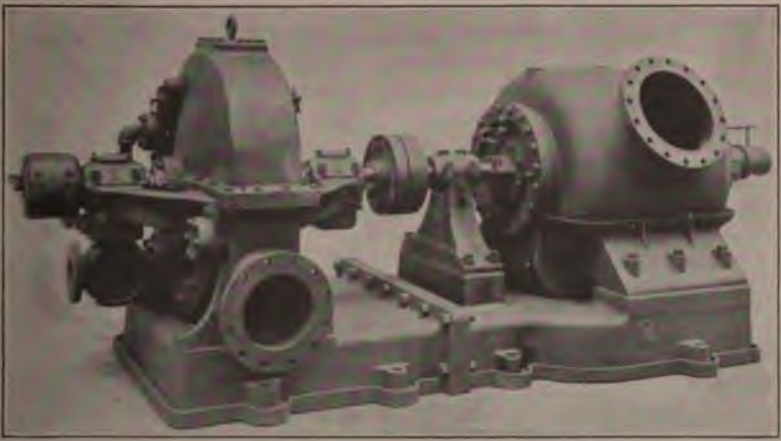


Fig. 146. A Trirotor Type of Steam-turbine-driven Pump for Circulating Water in Power Stations.

to handle 20,000,000 gallons of muddy river water per 24 hours at 1400 r.p.m. against a head of 130 to 160 feet, but at times as low as 95 feet.

Figs. 145 and 146 illustrate a birotor and a trirotor type of steam-turbine-driven pump for circulating water in power stations.

This application of steam-turbine drive to auxiliaries in power stations gives an economical installation and a maintenance cost less than for any other type.

CHAPTER XXIII.

COMMERCIAL PUMPS FOR GENERAL INDUSTRIAL USES.

It is the intention to here give a somewhat detailed description of the working parts of a standard type of pump. Many industries permit the use of such a type of pump, as, for instance, common contractors' work, irrigation, and similar work. Fig. 147 shows one of these commercial

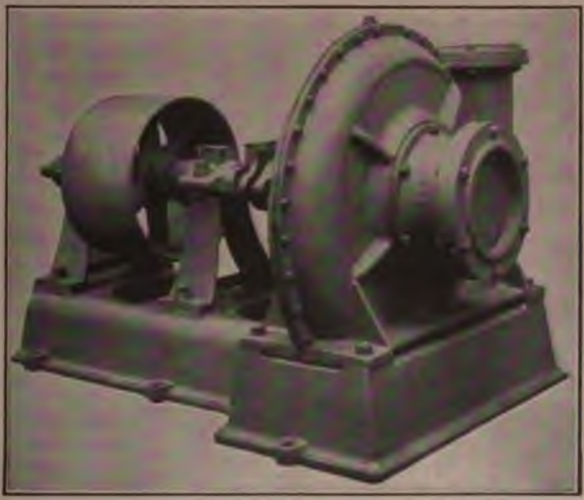


Fig. 147. Commercial Pump.

pumps having an average efficiency and a low first cost. For paper-mill, dye-house, and sugar-house work a special type of pump is required, in which the internal parts are easily accessible for cleaning without dismantling the entire machine. This style is shown in Fig. 148, and has its casing split along the horizontal line, as shown in Fig. 149. Such pumps are especially adapted for handling material which leaves a deposit inside of the pump, which must be regularly removed.

Another type, suitable for paper mills, sugar houses, and gas houses, is shown in Fig. 150, the casing and impeller being designed to handle heavy or thick liquors. They are suitable for pumping various kinds of liquors from chests or storage tanks to digesters and engines, for returning water

from the screens back to the filters, for pumping clay water from mixers to the beating engines, and lime used in bleaching and chloride of lime; in fact for every conceivable use in a mill.

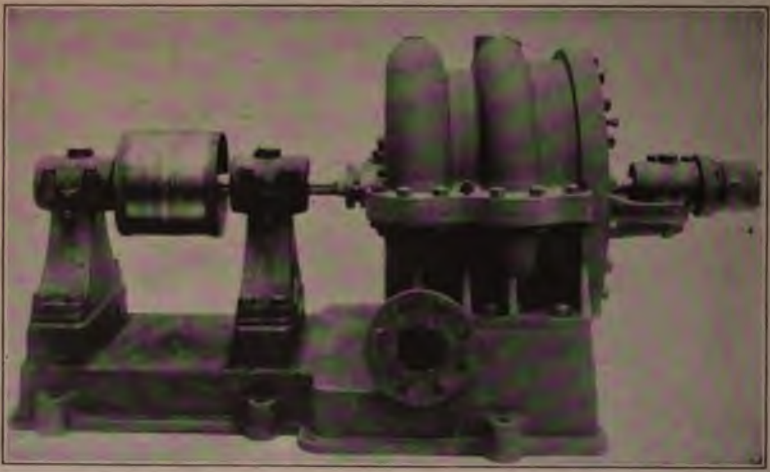


Fig. 148. Sugar-house Pump.

The material out of which such pumps should be made depends upon the liquid to be handled. Most of the trouble in such pumps can be attributed

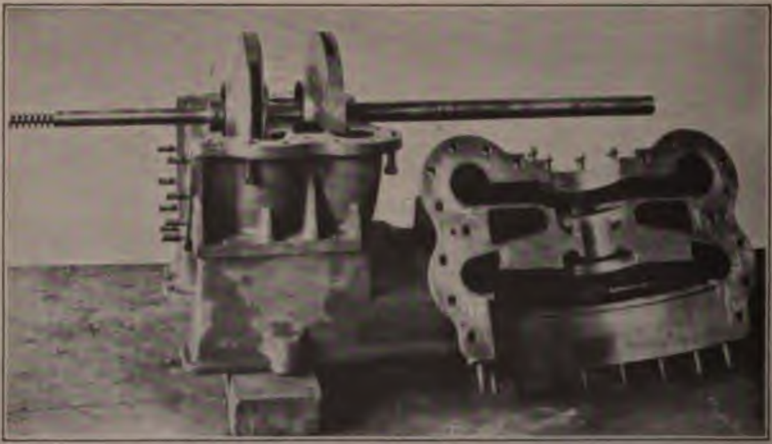


Fig. 149. Sugar-house Pump Showing Method of Examination.

to lack of knowledge of the liquid with which the pump is to be used, and if the conditions are known a pump can be made suitable to the particular service.

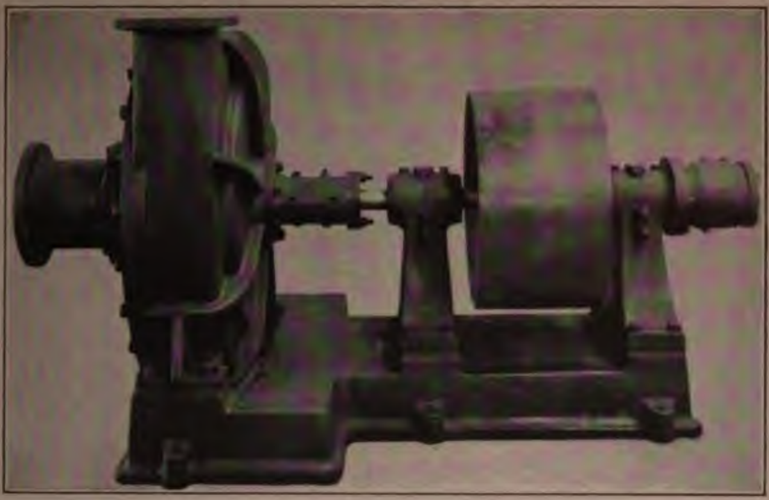


Fig. 150. Paper-mill Pump.

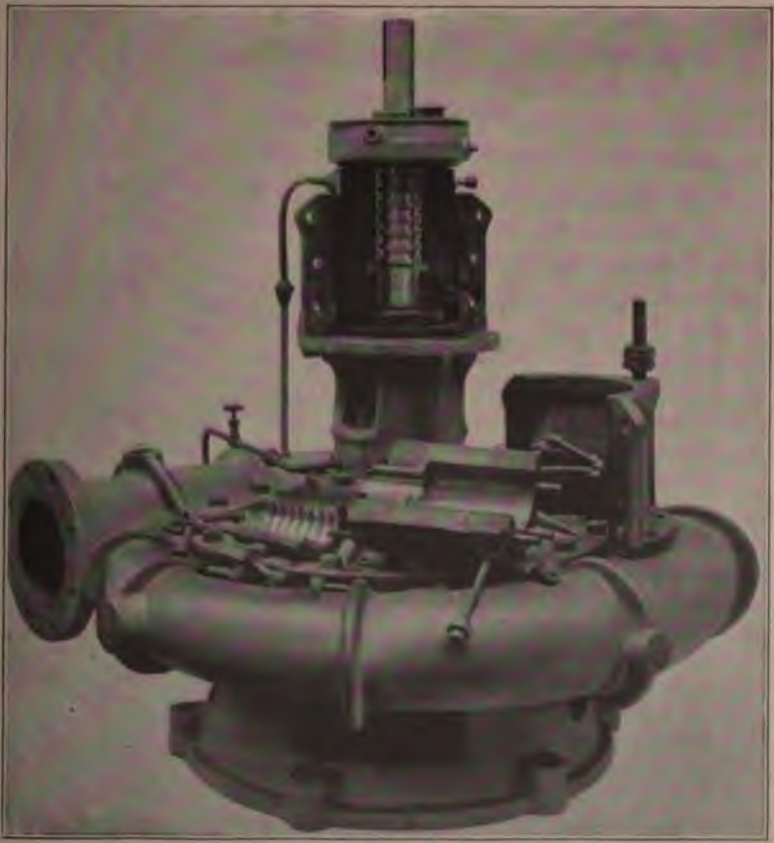


Fig. 151. Vertical Pump Showing Parts.

Where a vertical arrangement is necessary pumps like that shown in Fig. 151 may be used. A similar type for larger sizes and higher heads is shown in Fig. 152. In many modern buildings having deep basements, where water can accumulate, a type of sump pump, illustrated in Fig. 153, is used. This may be arranged to operate automatically, and as the pump is noiseless it is well adapted to buildings, apartment houses, and hotels. In small

Fig. 152. Vertical Turbine Pump.

units the turbine pump driven by electric motors with automatic float switches is being largely used in buildings for tank service.

To give the reader a fair idea of how the details of a modern centrifugal pump look, the following illustrations have been inserted. Fig. 154 shows the pump itself and the equipment usually needed. The casing may be of cast iron, cast steel, or bronze, according to nature of material to be handled. The inlet and outlet are shown in their normal position, but they are sometimes placed differently. Fig. 155 shows the impellers, which are also made of a material consistent with the properties of the liquid to be pumped. These impellers are balanced for high rotative speeds and may have either

Fig. 153. Vertical House Sump Pump.

a single side suction or a double suction inlet. They are designed according to mathematical formulas for each specific duty. The hubs or flanges shown are for the purpose of preventing leakage and for facilitating bal-

Fig. 154. Horizontal Turbine Pump, Vertical Split Casing.

ancing. In the spaces between the sides of the impellers and the casing walls there is an area which would be unbalanced, and, in order to overcome

Fig. 155. Arrangement of Impellers on a Multistage Pump.

this, hubs are provided of equal diameters on each side, having a running fit of not over $\frac{6}{1000}$ of an inch and having an end play of $\frac{1}{16}$ inch to allow

Fig. 156. Thrust Bearing.

the impeller to float to its proper position. Holes are drilled in the impellers at the middle, inside of these hubs, allowing water under the pressure to act upon the area within the rings, thus giving balance as near as is prac-

ticable. This arrangement is followed for each impeller, so that there is a minimum thrust, but to compensate for any possible thrust, whether due to

Fig. 157. Diffusion Vanes.

suction lift or to the wear that will take place in time, a self-oiling thrust bearing is supplied at the outer end as shown in Fig. 156. The additional

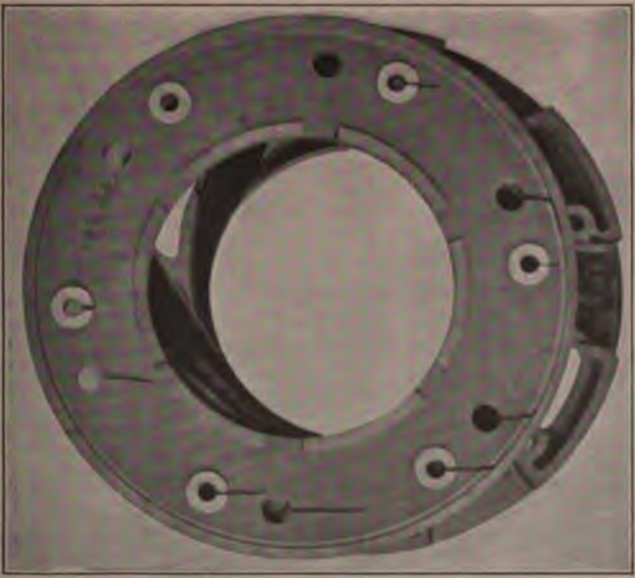


Fig. 158. Complete Diffusion.

purpose of the thrust bearing is to provide a lateral alignment between impellers and diffusion vanes. The main bearing is integral with the

stuffing box and secured to the casing by a recessed fitting. The stuffing box is sealed and of the lantern gland design, with soft packing each side of the gland. Fig. 157 illustrates the diffusion vanes, through which the water is guided in mathematically calculated passages, transforming the velocity head into pressure with the least possible losses. The material of these rings also depends upon the liquid. There are two rings, one with vanes and one without. Together they form the complete diffusion chamber shown in Fig. 158.

PART IV.—PRIME MOVERS FOR DRIVING CENTRIFUGAL PUMPS.

CHAPTER XXIV.

ELECTRIC MOTORS.

THE great majority of the centrifugal pumps in service are motor-driven. In many cases the importance of the relationship between the pump and the motor has been too little appreciated. In order to attain the best results, the engineer must fully understand the peculiarities of the pump, the variations in load during operations, the characteristics of electric power, and of the plant as a whole. The exact capacity and rating of the motor are of great importance, especially where power is bought on the motor rating. If the motor is too small, it will be constantly overloaded, and if it is too large, the customer pays for power not used. Furthermore, the power required at rated speed and head should not in itself determine the size of the motor. Maximum conditions should also be taken into account.

Having determined the size of motor required, it is of utmost importance that the proper style be selected for the limits of current available. It is in the selection of the class or style of motor that an understanding of the characteristics of the pump is absolutely essential, as it is necessary to select a motor which will take care of the varying loads and meet the different starting conditions. Where the head must vary, this may be accomplished by changing the speed, and a motor must be selected which permits speed regulation. The designer of the pump must, therefore, carefully consider the nature of his motor when laying out the characteristics of his impeller. On the other hand, the electrical engineer should design his motor to suit the characteristics of the pump.

A pump should be designed for an average head at maximum efficiency and maximum head at average efficiency, and motor and pump should be so proportioned that under the varying conditions the motor will not be overloaded except within a certain range, and will allow increased capacity for lower heads with nearly constant power. The pump should be designed for a wider range than is usual in order to give the motor a restricted output and to prevent excessive overloads under abnormally low heads. All pump characteristics should have a flat curve over a considerable range, making

the horse power nearly constant between the limits of the working conditions. This is especially necessary with induction motors, as both the power factor and the efficiency are reduced at light loads.

The problem of starting torque is easily solved with direct-current motors, provided their size is properly selected to correspond with the capacity and head of the pump under all conditions to be met. The majority, however, of pump installations are of the alternating-current type. An induction motor of proper capacity will start a pump successfully, provided the initial rush of current is tolerated. A synchronous motor may fail to start the pump, as the torque of this type of motor is usually small, and increases only as synchronism is reached; while the torque of the pump increases with the square of the velocity, thus producing a critical point in starting. Several remedies have been suggested, chiefly to start the pump empty and prime after full speed has been reached, also to start with a smaller independent motor to build up to speed. The load on the motor may be relieved in a primed pump by closing the discharge valve and allowing the pump to deliver water through a by-pass back to the suction under a lower head, removing a portion of the load from the motor. As the starting torque in a synchronous motor is due to eddy currents and hysteresis, more to the former than to the latter, it follows that every means of increasing the eddy currents will help the starting of the pump. The synchronous motor will hardly equal the induction motor for starting turbine pumps, as it has a large air gap and leakage and an uneven distribution of the secondary winding, which produces a lesser torque than can be found in an induction motor.

Another condition to be met with is the change of pressure by speed variation. In multipolar alternating-current motors this can be accomplished by using a different number of poles for the various speeds. The motor builder should know how the load on the pump varies with varying head and constant speed. Variation in the speed of direct-current motors is usually accomplished by resistance in the field, but this has a limited application and may prove objectionable for commutation, as the fields distort at the high speeds. The interpole variable-speed motor with a rheostat is better, although more expensive. In any case, the motor should be large enough so that the field rheostat cannot cut in too much resistance and overload the motor. The speed of the pumps should be properly selected under such conditions to avoid endless trouble due to the different speeds at different heads.

One must consider that in such combinations an increase of 10 per cent in speed may increase the power 50 per cent and cause a great increase in the armature losses. This is pointed out to show the great effect speed variation may have in an installation with a variable-speed motor. Speed is one of the most sensitive characteristics, and any unnecessary variation

is a great absorber of power. In a synchronous installation the speed is fixed by the prime mover and the motor is selected for maximum load. There are various methods for effecting the speed regulation. For example, a two-speed induction motor can operate at normal horse power on eight poles and at double the horse power on four poles. This is accomplished by primary windings connected to consecutive poles. As a four-pole motor the connections are in parallel, and for eight-pole in series. These motors can be designed and built for three speeds, operating on eight, ten, and sixteen poles, by having a primary with two sets of windings, one to eight-pole and sixteen-pole connections, and the other for ten-pole operations. This method of regulation is entirely practical and can be applied in connection with turbine pumps.

In vertical installations the motor should carry the weight of all revolving parts such as armature, impellers, and shafting, in order to obviate any vibration which may be set up. The pumps should be placed on good foundations. If this is not done the vibrations set up will ruin the motor or the pump.

Too much stress cannot be laid upon the fact that the pump and motor designers must work together and consider each other's problems more than has been done in the past; for the pump and motor must be considered together, just as a steam end in a direct-acting pump must be designed in connection with the pump end.

CHAPTER XXV.

STEAM ENGINES AND MISCELLANEOUS.

SINCE the development of high-speed steam engines, more engine-driven centrifugal pumps have been installed. This combination has met with

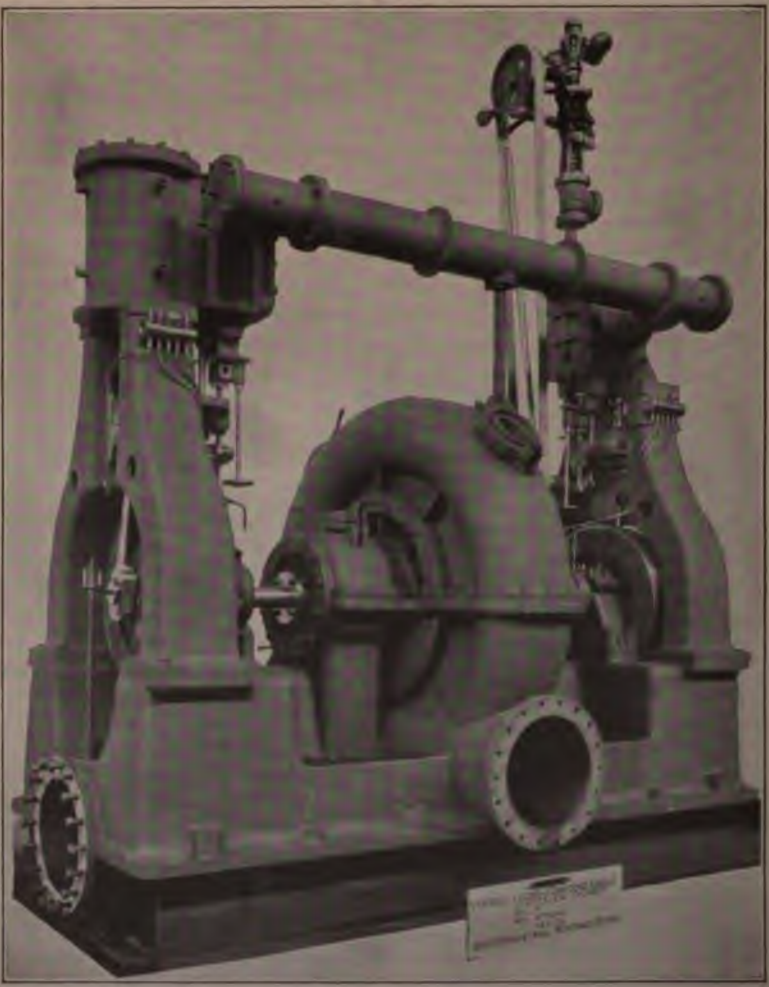


Fig. 159. Vertical Cross Compound Sewage Pump.

favor in small plants where the engineer in charge is better acquainted with the workings of a steam engine than with a motor or steam turbine. The speed of an engine-driven pump is necessarily limited. It rarely exceeds 800 r.p.m., and generally is in the neighborhood of 600 r.p.m. Such a unit cannot be used, therefore, for all kinds of service, and must be started slowly to give the engine sufficient time to warm up.

Gas and oil engines are also extensively used direct-connected to centrifugal pumps. One of the largest fields for this type is in contractors' work for emptying excavations, sumps, and ditches. Gasoline and kerosene-oil engines are used almost exclusively for this work, as they are convenient, portable, and need very little attention. Economy and efficiency are secondary considerations.

Centrifugal pumps for almost any kind of service may be driven by belts or silent chains if shaft power is available, as we need only consider the speed of the shafting and the ratio of the driver and driven pulleys to obtain the necessary speed.

Where there is an abundance of water under a few feet of head and a lesser quantity is desired at greater head, a combination centrifugal pump and water wheel has been found to be very economical. The water wheel or water turbine is direct-connected to a centrifugal pump which distributes the water under the desired head through a line of piping. This combination may also be used where there is an available water power for the water wheel, and a separate supply of pure water or some other liquid to be pumped.

Fig. 159 shows different types of engine-driven pumps, the former a compound engine with pump between, the latter the usual arrangement of single engine and pump. These types are commonly used for supplying circulating water for condenser, sewerage work, and similar duties.

CHAPTER XXVI.

STEAM TURBINES.

CENTRIFUGAL pumps, driven by steam turbines, are being extensively used for hot-well, boiler-feed, and circulating pumps. These combinations are efficient, occupy small space, and require minimum attendance. In the design of such a unit the chief problem is to construct turbines of 20 to 500 horse power which can operate at a speed suited to the pump. A compromise must be made between the ideal speeds of the pump and the turbine.

In large power plants, where steam can be used for heating feed water, the efficiency of an auxiliary prime mover, like a steam turbine, is of secondary importance, as low cost of operation will offset increased steam consumption. The rating of steam turbines in connection with pumps should always be on the maximum load, as otherwise the expected efficiency will not be obtained.

There are four principal types of steam turbines:

First. De Laval; an impulse turbine in which the steam is completely expanded in a single set of nozzles and all the kinetic energy is given up to a single row of blades.

Second. Parsons; impulse-reaction, where the energy of reaction of an expansion in the moving blades is added to the impulse of the steam as received from the fixed nozzles.

Third. Zoelly and Rateau; impulse turbine having a series of partial expansions, the energy of each expansion being absorbed in a single row of moving blades.

Fourth. Curtis; where the velocity of the steam from the nozzles is absorbed in and passes through several rows of moving blades.

All other types are modifications of these.

There are certain points towards which the effort of designers should be directed in order to secure the highest efficiency with maximum durability, simplicity, and cost of construction, namely:

First. Reduced steam consumption.

Second. Increased peripheral speed.

Third. Simplicity of design.

Fourth. Accuracy of workmanship.

Fifth. Provision for expansion and contraction under all conditions of load and steam pressure in a manner not to interfere with safe operation.

The small commercial turbines on the market to-day are the de Laval, Curtis, Terry, Kerr, Sturtevant, and Dake, all of the impulse type. The old-style de Laval has only one row of moving elements and one set of nozzles, necessitating high bucket speed; but in their recent designs there are several steam returns.

No steam turbine should be thought of that cannot ultimately meet the economy of the reciprocating engine, and small turbines under 300 horse power should have as near as possible the economy of the best grade of reciprocating engines, if the useful field is to be extended.

The following shows the effect of peripheral speeds on the economy of small turbines:

36-inch Curtis 19,000 feet per minute – 30 pounds per horse power per hour;
36-inch Curtis 25,000 feet per minute – 28 pounds per horse power per hour;
24-inch Terry 17,500 feet per minute – 40 pounds per horse power per hour;
24-inch Terry 10,000 feet per minute – 50 pounds per horse power per hour;
24-inch Kerr 17,500 feet per minute – 40 pounds per horse power per hour;
24-inch Kerr 6,000 feet per minute – 60 pounds per horse power per hour;
30-inch Bliss 20,000 feet per minute – 40 pounds per horse power per hour.

Single-stage turbines of the Electra type, of 45 horse power, running 3000 revolutions, with steam from 90 to 100 pounds, have given a steam rate of from 24 to 30 pounds per horse power condensing. This type is well adapted to small powers. It is known also by the name of Kolb, and is one of the best for use in connection with centrifugal pumps. It is entirely suitable for high-pressure steam. The only objection to it is the loss by friction in the guide blades, and the inability to use it for large powers, as there is not enough space on the periphery of the wheel for the necessary nozzles and guides.

A mixed-flow turbine or one using both high- and low-pressure steam is one that will be valuable in a great many places, particularly in steel mills, where an abundance of exhaust steam from the various rolling-mill engines, hammers, etc., may be utilized for running the pumps. Turbines can be built direct-connected with pumps, and arranged to be operated either by steam at atmospheric pressure and exhaust into a vacuum, or by high-pressure steam, exhausting into a vacuum or into the atmosphere. Such a turbine could be designed with two nozzle chambers, one for high-pressure steam and the other for low-pressure steam, having independent governor control.

The expansion of steam in a nozzle obtains a velocity of

$$V = 224\sqrt{H_1 - \{H_2X + q(1-X)\}},$$

where H_1 = total head at P_1;
H_2 = total head at P_2;
X = dryness fraction;
q = heat of liquid.

The available energy in steam between boiling point and absolute vacuum is 890,000 foot pounds per pound, and the velocity

$$V = \sqrt{2 \times 32.2 \times 890{,}000} = 7550 \text{ feet per second.}$$

This is based upon the theory advanced as to molecular velocity, and becomes important when the available energy in steam between different pressures is to be determined. The usual formula for determining this is: Foot pounds per pound of steam = $778\,[H_1 + C_p t_1 - (G_2 + Xv_2)]$.

H_1 = total heat at pressure p_1;
C_p = specific heat at superheated steam at pressure p_1;
t_1 = superheat in degrees Fahrenheit at pressure p_1;
G_2 = head of liquid at pressure p_2;
X = quality of steam at pressure p_2, or entropy;
v_1 = latent heat at pressure p_1;
v_2 = latent heat at pressure p_2.

Entropy of superheated steam can be calculated

$$C_p \log \frac{T_1 + t_1}{T_1} + \frac{v_1}{T_1} + \phi,$$

of moist steam,

$$\frac{Xv_2}{T_2} + \phi_2,$$

where T and T_2 absolute temperature of saturated steam at pressures p_1 and p_2, which equals 461 (temperature in Fahrenheit).

ϕ = entropy of water at pressure p_1;
ϕ_2 = entropy of water at pressure p_2.

From this can be found the moisture in per cent and available energy in foot pounds of the steam and the amount of moisture entering into the condensing apparatus, although it is to be taken into account that some of the moisture is lost. After the total energy is found by taking the efficiency of the turbine, the energy available can then be found.

The efficiency of a turbine is determined by the readings of pressure and temperatures and the amount of steam, exhaust pressure, and the electrical power of the generator, and is the theoretical water rate divided by the observed.

The small steam turbines now coming into general use vary from 10 horse power to 500 horse power. They are nearly all of the impulse type and promise to become the preëminent driving power for centrifugal or turbine pumps. The field for this combination is considerable and covers centrifugal pumps, feed pumps, condenser equipments, and marine auxiliaries. Steam consumption in some cases is of importance, but not in others.

Steam turbines of 200 to 500 horse power may be obtained having an economy equal to that of a reciprocating engine, and when the entire installation is considered the steam turbine will show an advantage.

APPENDIX.

ELECTRICAL DATA.

Full-load speeds for alternating-current motors based on 4 per cent slip.

TABLE NO. 8.

Number of poles.	Cycles.								
	25	27	30	33⅓	40	42	50	60	100
2	1440	1560	1730	1920	2300	2420	2880	....	....
4	720	780	865	960	1150	1210	1440	1725	2880
6	480	520	575	640	770	807	960	1150	1920
8	360	390	433	480	575	605	720	862	1440
10	290	310	345	385	460	485	575	690	1150
12	240	260	288	320	385	403	480	575	960
14	205	222	247	275	330	346	412	492	822
16	180	195	216	240	287	302	360	431	720
18	160	171	192	214	256	268	320	384	640
20	142	156	173	192	230	242	287	345	575

B.h.p. output for

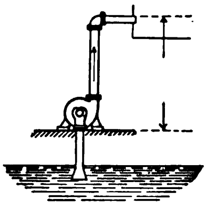
$$\text{alternating-current motor} = \frac{\text{volts} \times \text{ampere} \times \cos\phi \times \sqrt{n} \times m}{.746},$$

where n = number of phases;

$\cos\phi$ = power factor of motor;

m = motor efficiency.

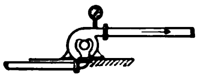
POINTS TO CONSIDER IN CENTRIFUGAL PUMP INSTALLATIONS.



Floor Line to Discharge Level (*Ft.*):

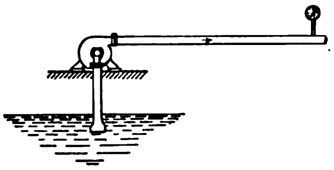
Should be given in open system. Pumping overboard or into a tank.

Mine Service: Give information about discharge line in addition to above.



Gauge Reading at Discharge Nozzle (*Lbs.*):

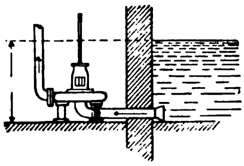
Should be given when pumping into a closed pressure system. (Heating System) or when pumping direct into city water main: Information about discharge line not needed.



Gauge Reading at End of Discharge Line (*Lbs.*):

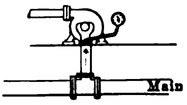
Should be given when a certain pressure is required at a distance away from the pump; for instance, in sprinkler systems and fire-hydrant systems. Information about discharge line is needed in addition to above.

Head-on Suction:



Floor Line to Suction Level (*Ft.*):

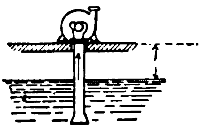
Should be given when suction water level is above floor line. This is the case with all submerged pumps, also when pump takes its water from an elevated tank, etc. Information about suction line is to be given in addition to above.



Gauge Reading at Suction Nozzle (*Lbs.*):

In closed system, when water enters under pressure; for instance, if connected to city main, in heating system, etc. Information about suction line not needed.

Suction Lift:



Floor Line to Suction Level (*Ft.*):

This is the most common case. Should always be given when pump takes its water from a well or from a river, provided pump is above water surface.

Information about suction pipe needed. State if foot valve is provided.

TABLE NO. 9.—PRESSURE OF WATER.

Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.	Feet head.	Pressure per square inch.
1	0.43	32	13.86	63	27.29	95	41.15	127	55.01	159	68.87	191	82.73	223	96.60	255	110.46	390	168.94
2	0.86	33	14.29	64	27.72	96	41.58	128	55.44	160	69.31	192	83.17	224	97.03	256	110.89	395	171.11
3	1.30	34	14.72	65	28.15	97	42.01	129	55.88	161	69.74	193	83.60	225	97.46	257	111.32	400	173.27
4	1.73	35	15.16	66	28.58	98	42.45	130	56.31	162	70.17	194	84.03	226	97.90	258	111.76	425	184.10
5	2.16	36	15.59	67	29.02	99	42.88	131	56.74	163	70.60	195	84.47	227	98.33	259	112.19	450	195.00
6	2.59	37	16.02	68	29.45	100	43.31	132	57.18	164	71.04	196	84.90	228	98.76	260	112.62	475	205.77
7	3.03	38	16.45	69	29.88	101	43.75	133	57.61	165	71.47	197	85.33	229	99.20	261	113.06	500	216.58
8	3.46	39	16.89	70	30.32	102	44.18	134	58.04	166	71.91	198	85.76	230	99.63	262	113.49	525	227.42
9	3.89	40	17.32	71	30.75	103	44.61	135	58.48	167	72.34	199	86.20	231	100.00	270	116.96	550	238.25
10	4.33	41	17.75	72	31.18	104	45.05	136	58.91	168	72.77	200	86.63	232	100.49	275	119.12	575	249.09
11	4.76	42	18.19	73	31.62	105	45.48	137	59.34	169	73.20	201	87.07	233	100.93	280	121.29	600	259.90
12	5.20	43	18.62	74	32.05	106	45.91	138	59.77	170	73.64	202	87.50	234	101.36	285	123.45	625	270.73
13	5.63	44	19.05	75	32.48	107	46.34	139	60.21	171	74.07	203	87.93	235	101.79	290	125.62	650	281.56
14	6.06	45	19.49	76	32.92	108	46.78	140	60.64	172	74.50	204	88.36	236	102.23	295	127.78	675	292.40
15	6.49	46	19.92	77	33.35	109	47.21	141	61.07	173	74.94	205	88.80	237	102.66	300	129.95	700	303.22
16	6.93	47	20.35	78	33.78	110	47.64	142	61.51	174	75.37	206	89.21	238	103.09	305	132.12	725	314.00
17	7.36	48	20.79	79	34.21	111	48.08	143	61.94	175	75.80	207	89.66	239	103.53	310	134.28	750	324.88
18	7.79	49	21.22	80	34.65	112	48.51	144	62.37	176	76.23	208	90.10	240	103.96	315	136.46	775	335.72
19	8.22	50	21.65	81	35.08	113	48.94	145	62.81	177	76.67	209	90.53	241	104.39	320	138.62	800	346.54
20	8.66	51	22.09	82	35.52	114	49.38	146	63.24	178	77.10	210	90.96	242	104.83	325	140.79	825	357.37
21	9.09	52	22.52	83	35.95	115	49.81	147	63.67	179	77.53	211	91.39	243	105.26	330	142.95	850	368.20
22	9.53	53	22.95	84	36.39	116	50.24	148	64.10	180	77.97	212	91.83	244	105.69	335	145.12	875	379.03
23	9.96	54	23.39	85	36.82	117	50.68	149	64.54	181	78.40	213	92.20	245	106.13	340	147.28	900	389.86
24	10.39	55	23.82	86	37.25	118	51.11	150	64.97	182	78.84	214	92.69	246	106.56	345	149.45	925	400.70
25	10.82	56	24.26	87	37.68	119	51.54	151	65.40	183	79.27	215	93.13	247	106.99	350	151.61	950	411.54
26	11.26	57	24.69	88	38.12	120	51.98	152	65.84	184	79.70	216	93.56	248	107.43	355	153.78	975	422.35
27	11.69	58	25.12	89	38.55	121	52.41	153	66.27	185	80.14	217	93.99	249	107.86	360	155.94	1000	433.18
28	12.12	59	25.55	90	38.93	122	52.84	154	66.70	186	80.57	218	94.43	250	108.29	365	158.10	1500	649.70
29	12.55	60	25.99	91	39.42	123	53.28	155	67.14	187	81.00	219	94.86	251	108.73	370	160.27	2000	866.30
30	12.99	61	26.42	92	39.85	124	53.71	156	67.57	188	81.43	220	95.30	252	109.16	375	162.45	3000	1299.50
31	13.42	62	26.85	93	40.28	125	54.15	157	68.00	189	81.87	221	95.73	253	109.59	380	164.61	......	....
.....	....	.....	....	94	40.72	126	54.58	158	68.43	190	82.30	222	96.16	254	110.03	385	166.78	......	....

TABLE NO. 10.—TABLE OF THEORETICAL HORSE POWER REQUIRED TO RAISE WATER TO DIFFERENT HEIGHTS.

Gallons per minute.	5 feet.	10 feet.	15 feet.	20 feet.	25 feet.	30 feet.	35 feet.	40 feet.	45 feet.	50 feet.	60 feet.	75 feet.	90 feet.	100 feet.	125 feet.	150 feet.	175 feet.	200 feet.	250 feet.	300 feet.	350 feet.	400 feet.	Gallons per minute.
5	.006	.012	.019	.025	.031	.037	.044	.05	.06	.06	.07	.09	.11	.12	.16	.19	.22	.25	.31	.37	.44	.50	5
10	.012	.025	.037	.050	.062	.075	.087	.10	.11	.12	.15	.19	.22	.25	.31	.37	.44	.50	.62	.75	.87	1.00	10
15	.019	.037	.056	.075	.094	.112	.131	.15	.17	.19	.22	.28	.34	.37	.47	.56	.66	.75	.94	1.12	1.31	1.50	15
20	.025	.050	.075	.100	.125	.150	.175	.20	.22	.25	.30	.37	.45	.50	.62	.75	.87	1.00	1.25	1.50	1.75	2.00	20
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37	.47	.56	.62	.78	.94	1.09	1.25	1.56	1.87	2.19	2.50	25
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.34	.45	.56	.67	.75	.94	1.12	1.31	1.50	1.87	2.25	2.62	3.00	30
35	.043	.087	.131	.175	.219	.262	.306	.35	.39	.44	.52	.66	.79	.87	1.08	1.31	1.53	1.75	2.19	2.62	3.06	3.50	35
40	.050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60	.75	.90	1.00	1.25	1.50	1.75	2.00	2.50	3.00	3.50	4.00	40
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67	.84	1.01	1.12	1.41	1.69	1.97	2.25	2.81	3.37	3.94	4.50	45
50	.062	.125	.187	.250	.312	.375	.437	.50	.56	.62	.75	.94	1.12	1.25	1.56	1.87	2.19	2.50	3.12	3.75	4.37	5.00	50
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90	1.12	1.35	1.50	1.87	2.25	2.62	3.00	3.75	4.50	5.25	6.00	60
75	.093	.187	.281	.375	.469	.562	.666	.75	.84	.94	1.12	1.40	1.69	1.87	2.34	2.81	3.28	3.75	4.69	5.62	6.56	7.50	75
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35	1.68	2.02	2.25	2.81	3.37	3.94	4.50	5.62	6.75	7.87	9.00	90
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50	1.87	2.25	2.50	3.12	3.75	4.37	5.00	6.25	7.50	8.75	10.00	100
125	.156	.312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87	2.34	2.81	3.12	3.91	4.69	5.47	6.25	7.81	9.37	10.94	12.50	125
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25	2.81	3.37	3.75	4.69	5.62	6.56	7.50	9.37	11.25	13.12	15.00	150
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62	3.28	3.94	4.37	5.47	6.56	7.66	8.75	10.94	13.12	15.31	17.50	175
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00	3.75	4.50	5.00	6.25	7.50	8.75	10.00	12.50	15.00	17.50	20.00	200
250	.312	.625	.937	1.250	1.562	1.875	2.187	2.50	2.81	3.12	3.75	4.69	5.62	6.25	7.81	9.37	10.94	12.50	15.72	18.75	21.87	25.00	250
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.40	5.62	6.75	7.50	9.37	11.25	13.12	15.00	18.75	22.50	26.25	30.00	300
350	.437	.815	1.312	1.750	2.187	2.625	3.062	3.50	3.94	4.37	5.25	6.56	7.87	8.75	10.94	13.12	15.31	17.50	21.87	26.25	30.62	35.00	350
400	.500	1.000	1.500	2.000	2.500	3.000	3.500	4.00	4.50	5.00	6.00	7.50	9.00	10.00	12.50	15.00	17.50	20.00	25.00	30.00	35.00	40.00	400
500	.625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50	9.37	11.25	12.50	15.62	18.75	21.87	25.00	31.25	37.50	43.75	50.00	500

This table gives the actual water horse power. Allowance must be made for friction losses in the machine and in the water pipes in estimating the size of steam end or motor required.

TABLE NO. 11.—CAPACITY IN GALLONS PER MINUTE AT VELOCITIES IN FEET PER SECOND, 1 TO 15 FEET; ALSO FRICTION HEAD IN FEET PER 100 FEET LENGTH OF PIPE.

Velocity.	¾-inch pipe.		1-inch pipe.		1¼-inch pipe.		1½-inch pipe.		2-inch pipe.		2½-inch pipe.		3-inch pipe.		4-inch pipe.		5-inch pipe.		Velocity.
	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	
1	1.50	1.19	2.90	1.04	4.50	.74	5.52	.54	11.0	.43	17.4	.31	24	.21	40	.12	68	.10	1
2	3.00	3.66	5.00	2.43	7.5	1.76	11.04	1.65	20.0	1.08	31.0	.76	47	.70	80	.48	128	.39	2
3	4.03	5.98	7.31	4.08	11.6	3.91	16.56	2.7	29.4	2.04	47.2	1.71	66	1.36	117	1.02	184	.82	3
4	5.50	10.49	9.82	6.83	15.1	5.79	22.0	4.5	39.2	3.4	61.0	2.6	81	2.28	156	1.71	245	1.67	4
5	6.89	13.64	12.24	10.25	19.4	8.32	27.4	6.8	48.9	5.1	76.5	4.1	110	3.4	196	2.56	306	2.05	5
6	8.25	19.12	14.68	14.33	23.2	12.34	32.9	9.5	58.7	7.2	91.8	5.7	132	4.7	236	3.6	367	2.87	6
7	9.65	25.44	17.13	19.08	26.8	16.56	34.4	12.7	68.5	9.5	107	7.6	154	6.3	275	4.8	428	3.8	7
8	11.01	32.64	19.58	24.50	30.3	20.82	43.8	16.3	78.3	12.2	122	9.8	176	8.1	314	6.1	490	4.9	8
8½	11.70	36.60	20.80	27.46	32.5	23.65	46.6	18.3	83.2	13.7	130	10.9	187	9.1	334	6.9	521	5.5	8½
9	12.39	40.40	22.03	30.58	34.5	25.51	49.4	20.4	88.1	15.2	138	12.2	198	10.1	352	7.6	551	6.1	9
9½	13.08	44.80	23.25	33.88	36.3	28.34	52.2	22.5	93.6	16.9	145	13.5	209	11.2	372	8.5	581	6.7	9½
10	13.79	49.68	24.48	37.33	38.3	30.53	55.3	24.9	97.9	18.6	153	14.9	220	12.4	391	9.3	612	7.5	10
10½	14.45	54.40	25.70	40.06	40.1	33.56	57.7	27.3	103	20.4	161	16.3	231	13.6	411	10.2	642	8.2	10½
11	15.15	59.60	26.92	44.75	42.6	36.41	60.5	29.8	108	22.3	168	17.9	242	14.9	431	11.1	673	8.9	11
11½	15.85	64.80	28.15	48.71	44.6	40.12	63.2	32.5	113	24.3	176	19.5	253	16.2	450	12.1	704	9.7	11½
12	16.52	70.40	29.37	52.83	46.8	43.73	66.6	35.2	117	26.4	184	21.0	264	17.6	471	13.2	735	10.6	12
13	17.94	82.00	31.82	61.58	49.5	49.91	71.5	41.0	127	30.7	199	24.6	286	20.5	510	15.3	796	12.3	13
14	19.30	94.80	34.27	71.00	53.5	57.25	77.8	47.3	139	35.5	214	28.4	308	23.7	549	17.7	857	14.2	14
15	20.75	108.00	36.72	81.08	57.4	64.82	82.5	54.0	149	40.5	230	32.4	330	27.0	587	20.3	918	16.2	15

TABLE NO. 12.—CAPACITY IN GALLONS PER MINUTE AT VELOCITIES IN FEET PER SECOND, 1 TO 15 FEET; ALSO FRICTION HEAD IN FEET PER 100 FEET LENGTH OF PIPE.—*Continued.*

Velocity.	6-inch pipe.		7-inch pipe.		8-inch pipe.		10-inch pipe.		12-inch pipe.		15-inch pipe.		18-inch pipe.		20-inch pipe.		24-inch pipe.		Velocity.
	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	
1	98	.09	118	.06	156	.05	248	.04	351	.03	551	.02	794	.03	963	.02	1,412	.02	1
2	176	.32	235	.28	312	.21	493	.18	704	.13	1101	.12	1,587	.11	1,956	.08	2,825	.07	2
3	264	.68	360	.58	470	.51	734	.41	1058	.34	1652	.27	2,379	.23	2,938	.20	4,230	.17	3
4	353	1.14	470	.98	627	.85	979	.68	1410	.57	2203	.46	3,173	.38	3,917	.34	5,640	.28	4
5	440	1.70	600	1.46	783	1.28	1224	1.02	1763	.87	2755	.68	3,965	.57	4,896	.51	7,051	.43	5
6	528	2.38	720	2.05	940	1.79	1468	1.43	2115	1.21	3304	.96	4,758	.80	5,875	.72	8,461	.60	6
7	617	3.18	840	2.72	1097	2.38	1713	1.90	2468	1.62	3855	1.27	5,552	1.06	6,854	.95	9,870	.79	7
8	706	4.08	959	3.49	1253	3.06	1958	2.45	2820	2.08	4407	1.63	6,345	1.36	7,833	1.22	11,280	1.01	8
8½	749	4.57	1019	3.92	1331	3.43	2081	2.74	2996	2.33	4688	1.82	6,742	1.52	8,323	1.37	11,985	1.14	8½
9	794	5.09	1079	4.36	1410	3.82	2203	3.05	3173	2.60	4958	2.04	7,138	1.70	8,812	1.53	12,690	1.27	9
9½	837	5.61	1139	4.83	1488	4.23	2326	3.38	3349	2.88	5232	2.25	7,535	1.88	9,302	1.69	13,395	1.40	9½
10	882	6.21	1199	5.33	1567	4.66	2448	3.73	3525	3.17	5508	2.50	7,931	2.07	9,792	1.87	14,100	1.55	10
10½	925	6.82	1259	5.84	1646	5.22	2571	4.09	3701	3.48	5783	2.73	8,329	2.27	10,281	2.05	14,805	1.70	10½
11	970	7.45	1319	6.39	1723	5.59	2693	4.47	3878	3.80	6058	2.98	8,725	2.48	10,771	2.24	15,510	1.86	11
11½	1013	8.11	1379	6.95	1801	6.08	2815	4.87	4054	4.14	6335	3.25	9,122	2.70	11,258	2.43	16,215	2.03	11½
12	1058	8.80	1439	7.54	1880	6.60	2938	5.28	4230	4.49	6610	3.52	9,518	2.93	11,750	2.64	16,920	2.20	12
13	1145	10.26	1559	8.74	2037	7.90	3128	6.15	4583	5.23	7161	4.10	10,310	3.42	12,729	3.08	18,330	2.56	13
14	1233	11.83	1679	10.14	2193	8.87	3472	7.10	4935	6.03	7711	4.73	11,104	3.93	13,708	3.55	19,740	2.95	14
15	1321	13.51	1799	11.58	2350	10.01	3672	8.10	5288	6.89	8262	5.40	11,897	4.50	14,688	4.05	21,150	3.37	15

TABLE NO. 13.—CAPACITY IN GALLONS PER MINUTE AT VELOCITIES IN FEET PER SECOND, 1 to 15 FEET; ALSO FRICTION HEAD IN FEET PER 100 FEET LENGTH OF PIPE.—*Continued.*

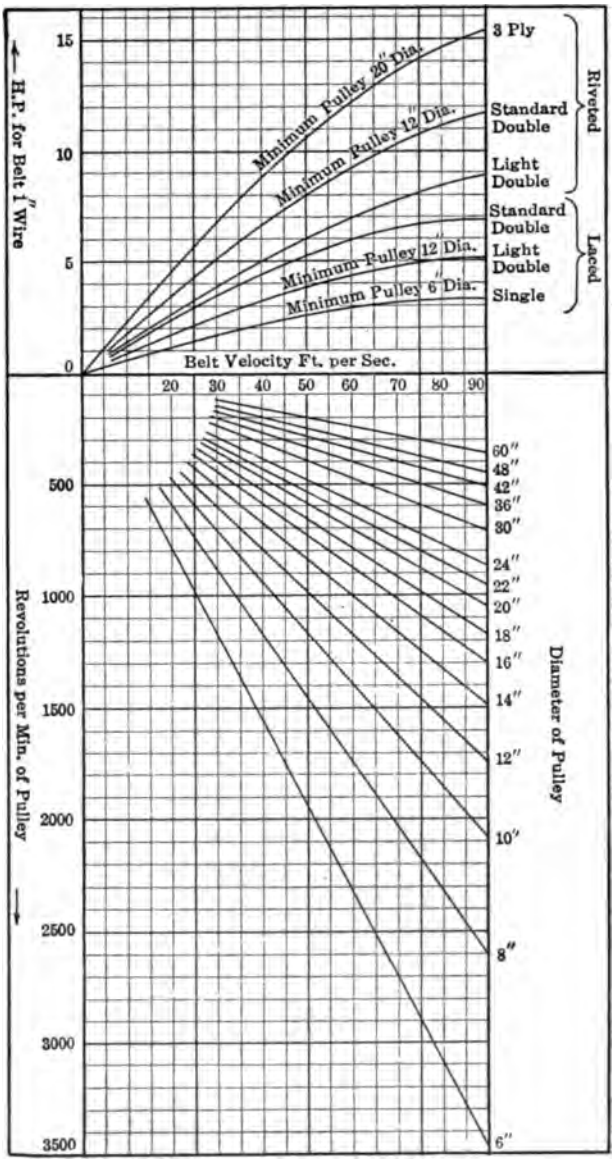
Velocity.	30-inch pipe.		32-inch pipe.		36-inch pipe.		42-inch pipe.		45-inch pipe.		48-inch pipe.		54-inch pipe.		60-inch pipe.		72-inch pipe.		Velocity.
	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction.	Capacity.	Friction	Capacity.	Friction.	Capacity.	Friction.	
1	2,200	.013	2,507	.013	3,120	.012	4,319	.010	4,958	.009	5,700	.008	7,138	.006	8,800	.005	12,690	.004	1
2	4,400	.051	5,014	.048	6,240	.043	8,637	.039	9,915	.035	11,280	.031	14,277	.028	17,600	.025	25,381	.021	2
3	6,610	.136	7,520	.128	9,518	.113	12,955	.097	14,872	.091	16,921	.085	21,415	.076	26,438	.068	38,071	.057	3
4	8,813	.228	10,027	.214	12,690	.190	17,273	.163	19,829	.152	22,561	.145	28,553	.127	35,251	.114	50,762	.095	4
5	11,015	.341	12,534	.320	15,863	.284	21,591	.244	24,786	.228	28,201	.214	35,692	.190	44,064	.171	63,452	.142	5
6	13,218	.478	15,040	.448	19,036	.397	25,910	.341	29,743	.319	33,845	.299	42,830	.265	52,877	.239	76,142	.199	6
7	15,421	.636	17,547	.596	22,208	.528	30,228	.454	34,700	.424	39,481	.398	49,968	.353	61,689	.318	88,833	.265	7
8	17,624	.816	20,054	.765	25,381	.679	34,546	.583	39,657	.544	45,121	.510	57,107	.454	70,502	.408	101,523	.340	8
8½	18,725	.915	21,307	.858	26,967	.760	36,705	.654	42,136	.610	47,941	.572	60,676	.508	74,909	.458	107,868	.381	8½
9	19,827	1.01	22,561	.956	28,554	.847	38,864	.728	44,615	.680	50,762	.637	64,245	.566	79,315	.510	114,214	.425	9
9½	20,928	1.12	23,814	1.06	30,140	.938	41,023	.807	47,093	.753	53,582	.706	67,814	.627	83,721	.565	120,559	.470	9½
10	22,030	1.24	25,067	1.17	31,726	1.03	43,183	.899	49,572	.830	56,402	.778	71,383	.691	88,128	.622	126,904	.519	10
10½	23,131	1.36	26,321	1.28	33,313	1.13	45,342	.975	52,050	.910	59,222	.853	74,953	.758	92,534	.683	133,249	.569	10½
11	24,233	1.49	27,574	1.40	34,899	1.16	47,501	1.07	54,529	.994	62,042	.932	78,522	.829	96,941	.746	139,594	.622	11
11½	25,338	1.62	28,828	1.52	36,485	1.35	49,660	1.16	57,208	1.08	64,862	1.02	82,091	.902	101,347	.812	145,940	.677	11½
12	26,436	1.76	30,081	1.65	38,072	1.46	51,819	1.26	59,486	1.17	67,682	1.10	85,660	.978	105,753	.881	152,285	.734	12
13	28,639	2.05	32,588	1.92	41,214	1.70	56,137	1.47	64,443	1.37	73,322	1.28	92,798	1.14	114,566	1.03	164,975	.855	13
14	30,842	2.37	35,094	2.22	44,417	1.97	60,456	1.69	69,408	1.58	78,962	1.48	99,937	1.32	123,379	1.18	177,666	.986	14
15	33,045	2.70	37,601	2.53	46,590	2.24	64,774	1.93	74,358	1.80	84,603	1.69	107,075	1.50	132,192	1.35	190,356	1.23	15

TABLE NO. 14.—TABLES OF DRIVING POWER OF LEATHER BELTS FOR CENTRIFUGAL PUMPS.

Diameter of pump pulley.		Revolutions per minute.	100	150	200	250	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600	1800	2000	2250	2500	2750	3000	3250	3500	3750	4000
Ft.	In.	B.h.p. Transmitted by each inch in breadth of single belt.																												
0	4																								1.4	1.6	1.8	2.0	2.3	2.5
0	4½																							1.6	1.8	2.1	2.4	2.6	2.8	3.0
0	5																						1.8	2.0	2.3	2.6	2.9	3.2		
0	5½																					1.9	2.2	2.6	3.0	3.3	3.6			
0	6																				2.0	2.3	2.7	3.1	3.6	3.9				
0	7																			2.8	3.3	3.9	4.4	5.1						
0	8																2.7	3.0	3.3	3.8	4.3	5.0							
0	9															3.0	3.3	3.6	0.4	4.7	5.4	6.3							
0	10													2.9	3.2	3.6	4.0	4.4	4.85	5.7	6.5								
0	11												3.0	3.4	3.9	4.3	4.7	5.2	5.7	6.7									
1	0											3.1	4.0	4.5	5.0	5.5	6.0	6.8											
1	2										3.5	4.1	4.7	5.4	6.0	6.7	7.4												
1	4									3.5	4.6	5.4	6.2	7.0	8.1														
1	6								3.5	4.9	5.8	6.8	7.8	8.9															
1	8							3.3	4.8	6.0	7.3	8.5	10.0																
1	10							4.4	5.6	6.9	8.4	10.0																	
2	0						4.2	5.08	6.4	7.9	9.6																		
2	6					3.9	5.45	7.3	9.45	11.4																			
3	0				4.0	5.0	7.0	9.5	12.5																				
3	6			3.82	5.1	6.3	9.5	12.2																					
4	0			4.46	6.0	7.6	11.1																						
5	0		4.5	6.48	8.5	13.8																							
6	0		5.3	8.6	11.9	14.8																							
7	0		4.5	7.3	11.0	14.0																								
8	0		5.8	9.27	13.3																									
9	0																													
10	0																													

REMARKS. — The values given in this table must not be exceeded. The lower side of belt to be tight. Ratio of pulley diameters, maximum 1 : 5. Distance of centers for small belts (up to 4 inches wide), 15 feet; for large belts, 30 feet. Best thickness of belt: for 4-inch pulley, ⅛-inch; for 6-inch pulley, $\frac{3}{16}$-inch, etc. For unfavorable conditions [(a) transmission to lower speed; (b) crossed belts; (c) small distance of centers, etc.] the values given must be reduced.

Example.—To obtain size of pump pulley and belt see belt curve, page 4; and for 742 revolutions per minute find diameter of pulley, assum-



ing best belt velocity at about 75 feet per second. Curve points to 22-inch diameter pulley, belt velocity 71 feet per second.

On the curve for standard riveted double belt the horse power per inch of width for 71 feet per second belt velocity = 10.3 horse power. There-

fore for transmitting 84.7 horse power the width of belt required is $84.7 \div 10.3 = 8.22$ inches. Take a belt 9 inches wide and a pulley with a 10-inch face.

$$\text{Driving pulley} = \frac{742 \times 22}{250} = 65.3 \text{ inches, say 66-inch diameter pulley,}$$

which allows for 1 per cent slip.

TABLE NO. 15.—SPEED OF ROTARY FIELD FOR DIFFERENT NUMBERS OF POLES AND FOR VARIOUS FREQUENCIES.

Number of poles.	Speed of revolving magnetism, in revolutions per minute, when frequency is:											
	25	30	33⅓	40	50	60	66⅔	80	100	120	125	133⅓
2	1500	1870	2000	2400	3000	3600	4000	4800	6000	7200	7500	8000
4	750	900	1000	1200	1500	1800	2000	2400	3000	3600	3750	4000
6	500	600	667	800	1000	1200	1333	1600	2000	2400	2500	2667
8	375	450	500	600	750	900	1000	1200	1500	1800	1875	2000
10	302	360	400	480	600	720	800	960	1200	1440	1500	1600
12	250	300	333	400	500	600	667	800	1000	1200	1250	1333
14	214	257	286	343	428	514	571	686	857	1029	1071	1143
16	188	225	250	300	375	450	500	600	750	900	938	1000
18	167	200	222	267	333	400	444	533	667	800	833	889
20	150	180	200	240	300	360	400	480	600	720	750	800
22	136	164	182	217	273	327	364	436	545	655	682	720
24	125	150	167	200	250	300	333	400	500	600	625	667

TABLE NO. 16.—SLIP OF INDUCTION MOTORS.

Capacity of motor, H.P.	Slip, at full load, per cent.		Capacity of motor, H.P.	Slip, at full load, per cent	
	Usual limits.	Average.		Usual limits.	Average.
⅛	20–40	30	15	5–11	8
¼	10–30	20	20	4–10	7
½	10–20	15	30	3–9	6
1	8–20	14	50	2–8	5
2	8–18	13	75	1–7	4
3	8–16	12	100	1–6	3.5
5	7–15	11	150	1–5	3
7½	6–14	10	200	1–4	2.5
10	6–12	9	300	1–3	2

FRICTION OF WATER IN PIPES.

The following tables show the flow of water in pipes 1000 feet long, as computed by the Hazen-Williams formula $V = 1.32\,Cr^{.63}\,S^{.54}$,

Where V = velocity of water in feet per second.

C = an empirical coefficient.

r = the hydraulic radius in feet $= \frac{d}{4}$ for pipes running full.

S = the hydraulic slope $= \frac{h}{l} = \frac{head}{1000'}$.

The friction heads are given for $C = 100$ only.

For other values of C multiply by $A = \sqrt[.54]{\frac{100}{C}}$.

TABLE NO. 17.—FRICTION HEAD IN FEET PER 1000 FEET LENGTH OF PIPE.

NOTE. — Figures in this Table give the friction head in feet per 1000 feet length of pipe based on factor C = 100 for pipes of condition or age represented opposite factor C = 100 in Table 3.

$C = 100$	Size of pipe in inches. Standard wrought-iron pipe. ⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	4	5
Inside dia. d =	.27	.364	.494	.623	.824	1.048	1.38	1.611					
Approximate. At const. head cap. is prop. to $d^{2.5}$ =	0378	.08	172	.306	.616	1.12	2.24	3 3	5.7	9.9	15.6	32	56
Approximate. At const. capacity head is prop. to $\frac{1000}{d^5}$ =	700,000	157,000	34,000	10,700	2,630	790	200	92.5	31.25	10.2	4.12	.977	32
U. S. gallons per minute = G.													
.2	62												
.5	345	78											
1	1210	280	64	21									
1.5		600	138	45									
2		1030	233	74	19								
3			490	158	41	12.6							
4			840	270	70	21.4	5.7	2.62					
5			1260	410	105	32.5	8.4	3.98					
6				570	147	45.5	12	5.6	2				
8				980	250	78	20.3	9.5	3.3	1.1			
10				1470	380	117	30.5	14.3	5	1.7	.7		
15					800	250	65	30	10.7	3.6	1.5		
20					1360	420	111	52	18.2	6.1	2.5	.62	
30						800	235	110	38.4	12.9	5.4	1.32	.44
40						1520	400	188	66	22	9.1	2.23	.75
50							600	284	99	33.2	13.8	3.39	1.13
60							850	396	139	46.5	19.2	4.72	1.59
80							1450	680	237	79	32.8	8.1	2.71
100								1020	358	120	49.6	12.2	4.11
150									760	256	105	26	8.7
200									1290	431	178	44	14.8
300										920	380	93	31.4
400										1560	650	160	54
500											980	240	81
600												337	113

For other values of C multiply figures in Table by A.

At constant diameter head is approximately proportional to G^2.

TABLE NO. 18.—TABLE GIVING COEFFICIENT OF CONDITION C, FACTOR a AND CORRESPONDING YEARS OF SERVICE IN SOFT, CLEAR, UNFILTERED RIVER WATER.

C	$\sqrt[.54]{\frac{100}{C}} = A$	Condition of pipe. ¼ – 1½	Condition of pipe. 2 – 3	4	5	6	8	10	12	16	20	24	30	36	42 48	54	60
				Years of service of cast-iron pipe.													
140	.54	Very smooth and straight W. I.	Do. brass, tin, etc.	00	00	00	00	00	00	00	00	00	00	00	00	00	00
130	.62	....	Ordinary straight brass, tin, etc.	0	0	0	0	0	0	0	0	0	0	0	0	0	0
120	.71	Smooth new wrought iron		4	4	4	5	5	5	5	5	5	6	6	6	6	6
110	.84	....	....	...	...	...	10	10	10	11	11	11	12	12	12	12	12
100	**1**	**Ordinary Wrought Iron**		**13**	**14**	**15**	**16**	**17**	**17**	**18**	**19**	**19**	**19**	**20**	**20**	**20**	**20**
90	1.22	....	....	...	...	...	...	...	26	27	28	29	30	30	30	31	31
80	1.52	Old wrought iron		26	28	30	33	35	37	39	41	42	43	44	45	46	47
60	2.58	....	Very rough	45	50	55	62	68	...	...	...	...	...	...	...	...	...
40	5.46	Very rough	Badly tuberculated	75	87	95	...	...	...	...	...	...	...	...	...	...	...

Diameter of pipe in inches.

00 means Very Best and New Cast Iron Straight Pipe. 0 means Good New Cast Iron Pipe.

TABLE NO. 19.—FRICTION HEAD IN FEET PER 1000 FEET LENGTH OF PIPE.

NOTE.—Figures in this Table give the friction head in feet per 1000 feet length of pipe based on factor $C = 100$ for pipes of age represented opposite factor $C = 100$ in Table 3.

$C = 100$	6	8	10	12	16	20	24	30	36	42	48	54	60
Approximate. At const. head cap. is prop. to $d^{2.5}$ =	88.2	181	316	499	1024	1790	2820	4930	7780	11,430	16,000	21,400	27,900
Approximate. At const. capacity head is prop. to $\frac{100 \text{ Mill}}{d^5}$ =	12,900	3,050	1,000	402	954	·31.25	12.6	4.12	1.65	.765	.403	.218	.129
U. S. Gallons per minute = G. 35	.24												
42	.33												
56	.57	.13											
70	.86	.22	.07	.03									
105	1.84	.43	.16	.07									
140	3.1	.77	.26	.11	.026								
210	6.6	1.62	.55	.22	.056								
280	11.3	2.76	.93	.38	.095	.032							
350	16.9	4.18	1.41	.58	.143	.049	.02						
420	23.8	5.9	1.97	.81	.201	.068	.03						
560	40.4	9.9	3.38	1.38	.34	.115	.05						
700	61	15.1	5.1	2.1	.52	.174	.072	.024					
1,400	222	54	18.4	7.6	1.87	.63	.259	.087	.036				
2,100	......	116	38.6	16.0	3.98	1.33	.55	.184	.076	.036			
2,800	......	......	66	27	6.8	2.28	.93	.315	.129	.061	.032	.018	.011
3,500	......	......	......	41	10.2	3.43	1.41	.476	.196	.092	.048	.028	.016
4,200	......	......	......	58	14.3	4.81	1.97	.67	.274	.129	.068	.038	.023
5,600	......	......	......	99	24.2	8.2	3 38	1.13	.467	.22	.115	.065	.039
7,000	......	......	......	150	36.8	12.4	5.1	1.72	.71	.332	.174	.098	.059
10,500	......	......	......	......	78	26.2	10 8	3.64	1.49	.7	.37	.21	.125
14,000	......	......	......	......	......	44.8	18.3	6.2	2.55	1.21	.63	.354	.212
21,000	......	......	......	......	......	......	39	13.2	5.4	2.56	1.33	.75	.449
28,000	......	......	......	......	......	......	......	22.4	9.2	4.35	2.28	1.28	.76
35,000	......	......	......	......	......	......	......	......	13.9	6.6	3.44	1.94	1.16
42,000	......	......	......	......	......	......	......	......	19.6	9.2	4.8	2.71	1.62
56,000	......	......	......	......	......	......	......	......	......	15.7	8.2	4.61	2.78
70,000	......	......	......	......	......	......	......	......	......	......	12.4	7	4.19

Size of pipe in inches.

For other values of C multiply figures in Table by A.

At constant diameter head is approximately proportional to G^2 and capacity is approximately proportional to $\sqrt{h}$.

APPROXIMATE RULES FOR INTERPOLATING.

RULE 1. — *At constant head, capacity is proportional to* $d^{2.5}$.

Example: A 4-inch pipe discharges 60 gallons, how much would a 1-inch pipe discharge under the same conditions?

$$4^{2.5} = 32, \quad 1.048^{2.5} = 1.12, \text{ hence capacity} = \frac{60 \times 1.12}{32} = 2.1 \text{ g.p.m.}$$

RULE 2. — *At constant capacity, head is proportional to* $\frac{1}{d^5}$.

Example: Capacity 600 g.p.m., diameter = 3 inches. What is the head? The nearest figure in Table is 337 feet for a 4-inch pipe, and the friction in a 3-inch pipe must be *greater*, hence $h = \frac{337 \times 4.12}{.977} = 1420$ feet.

RULE 3. — *At constant diameter, head is proportional to* G^2.

Example: Same as before.

The nearest figure in table is 980 feet head for 500 g.p.m., and the friction for 600 g.p.m. must be *greater*, hence $h = \frac{980 \times 600^2}{500^2} = 1410$ feet.

RULE 4. — *At constant diameter, capacity is proportional to* $\sqrt{h}$.

Example: Diameter 20 inches, h = 64 feet per 1000 feet length. What is the capacity?

The nearest figure in Table is 44.8 feet giving 14,000 g.p.m. The capacity will be *greater*, hence $\frac{14{,}000 \times \sqrt{64}}{\sqrt{44.8}} = \frac{14{,}000 \times 8}{6.7} = 16{,}700$ g.p.m.

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